

# **Development of Flow Work Exchangers for Energy Recovery in Reverse Osmosis Plants**

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James R. Smith, Assistant Secretary for Water and Power Resources**

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## FOREWORD

This is one of a continuing series of reports designed to present accounts of progress in saline water conversion and the economics of its application. Such data are expected to contribute to the long-range development of economical processes applicable to low-cost demineralization of sea and other saline water.

Except for minor editing, the data herein are as contained in a report submitted by the contractor. The data and conclusions given in the report are essentially those of the contractor and are not necessarily endorsed by the Department of the Interior.

# TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
	LIST OF ILLUSTRATIONS	iv
	LIST OF TABLES	v
	NOMENCLATURE	vi
	HYDRAULIC SYMBOLS	vii
	SUMMARY	viii
1	INTRODUCTION	1
2	PROGRAM OBJECTIVES AND TASKS	4
3	FLOW WORK EXCHANGER PERFORMANCE EQUATIONS	6
4	TEST PROGRAM ON TWO PROTOTYPE FLOW WORK EXCHANGERS	8
	4.1 Work Exchanger Configurations and Modifications	8
	4.2 Start-up and Operation Procedures	11
	4.3 Test Program and Results	17
	4.4 Test Observations	22
	4.5 Test Program Conclusions	26
5	DESIGN OF CONTROLS FOR AUTOMATIC OPERATION OF A WORK EXCHANGER	28
	5.1 Background	28
	5.2 Objectives for the Automatic Control System	28
	5.3 Control System Design Considerations	29
	5.4 Alternate Feed Pump Arrangements	32
	5.5 Control System Design	34
6	COMPARISON OF BLADDER AND PISTON TYPE FLOW WORK EXCHANGERS	38
	6.1 Objectives and Scope of Comparison	38
	6.2 Factors Considered in the Comparison	38
	6.2.1 Potential for System Scale-Up	38
	6.2.2 Reliability of Accumulators and Valves	39
	6.2.3 Efficiency of Flow Work Exchangers	40
	6.2.4 Comparative System Costs	42
	6.3 Cost Effective Analysis - Summary and Conclusion	43

## TABLE OF CONTENTS (Concluded)

<u>Section</u>		<u>Page</u>
7	RELIABILITY TESTING PROGRAM OF A PISTON TYPE WORK EXCHANGER	46
7.1	Experimental Observations	47
7.2	Discussion of Specific Components	47
7.3	Conclusions	54
8	CONCLUSIONS	56
9	RECOMMENDATIONS	58
	BIBLIOGRAPHY	60
<u>Appendix</u>		
A	DERIVATION OF PERFORMANCE EQUATIONS FOR FLOW WORK EXCHANGERS	61
A.1	Ideal Energy Exchange Between Two Liquid Streams	61
A.2	Losses in an Actual Flow Work Exchange System	63
A.3	System Efficiency Losses Due to Leakage of High- Pressure Flow	67
B	DETAILED EVALUATION OF COST AND EFFICIENCY FOR A RANGE OF FLOW WORK EXCHANGER SIZES	69
B.1	Description of Approach	69
B.2	Component Pressure Loss Information	72
B.3	Component Cost Data	75
B.4	Pump Efficiency Data	79
B.5	Selection and Evaluation of 18 Flow Exchanger Systems	82

## LIST OF ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
4-1	Original Flow Work Exchanger Test System Schematic	10
4-2	Modified Flow Work Exchanger Test System Schematic	12
4-3	Accumulator and Piston Configurations	13
4-4	Picture of Modified Piston Flow Work Exchanger (Front View)	14
4-5	Picture of Modified Piston Flow Work Exchanger (Side View)	15
4-6	Measured Pressure Losses for Bladder and Piston Type Flow Work Exchangers	19
4-7	Calculated Work Exchanger Efficiencies Based upon Measured Pressure Losses	20
5-1	Schematic Arrangement of the Work Exchanger	30
5-2	Low-Pressure Side of the Work Exchanger Flow Path	30
5-3	Pump Operation with the Simple Control System	31
5-4A	Feed Pump for Work Exchanger Only	33
5-4B	Feed Pump Serves Complete System	33
5-5	Work Exchanger Control System	35
6-1	Piston Type Flow Work Exchanger Cost and Efficiency Vs. Flow Rate	41
7-1	Polypak Piston and Seal	52
A-1	Ideal Indicator Diagram for a Flow Work Exchanger	62
A-2	Indicator Diagram for Actual Flow Work Exchanger System	62
B-1	Standard Flow Work Exchanger for Cost Effective Analysis	70
B-2	Ball Valve Pressure Loss	73
B-3	Check Valve Pressure Loss	74
B-4	Piston Accumulator Entrance and Discharge Losses Due to Expansions and Contractions	76
B-5	Pipe and Elbow Pressure Loss	77
B-6	Cost of Accumulators, Ball Valves, and Check Valves	78
B-7	Flow Work Exchanger Pump Costs	80
B-8	Pump Efficiency Data	81
B-9	Increased Pump Cost and Decreased Flow Work Exchanger Efficiency Due to Increased System Pressure Loss	87

## LIST OF TABLES

<u>Table</u>		<u>Page</u>
4-1	SUMMARY OF THE PROGRAM	18
4-2	SUMMARY OF SYSTEM FLOW AND PRESSURE FLUCTUATIONS DURING VALVE SWITCHING	24
5-1	CONTROL SYSTEM COMPONENTS	37
6-1	COMPARISON OF FLOW WORK EXCHANGE UNITS	44
7-1	RESULTS OF WORK EXCHANGER RELIABILITY TESTING	48
A-1	FLOW WORK EXCHANGER SYSTEM LOSSES	64
B-1	COMPONENTS FOR STANDARD SYSTEM IN FIGURE B-1	71
B-2	TABULATION OF COMPONENT SIZES AND COSTS FOR COST EFFECTIVE ANALYSIS	84
B-3	FLOW WORK EXCHANGER EFFICIENCY	85

## NOMENCLATURE

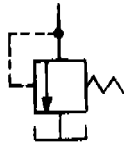
A	-	in <sup>2</sup>	-	Area
C	-	fps	-	Velocity
F	-		-	Fraction of cycle where pressure and flow transients occur during valve switching or bottoming of piston
H	-	ft	-	Head rise in pumps, or Head loss in components
m	-	lb/sec	-	Average air flow rate for pneumatic valves
N	-	cpm	-	Accumulator cycling rate
P	-	psig	-	Pressure
Q	-	gpm	-	Liquid flow rate
V	-	ft <sup>3</sup>	-	Accumulator operating displacement.
W	-	$\frac{\text{ft.lb.}}{\text{sec.}}$	-	Energy exchange or loss
$\Delta h$	-	Btu	-	Air enthalpy rise for pneumatic air supply system.
$\Delta P$	-	psid	-	Measured pressure differential.

### Subscripts

a	-	pneumatic system
b	-	brine
c	-	controls
co	-	contraction loss
e	-	expansion loss
f	-	feed
h	-	high pressure
i	-	ideal
l	-	low pressure
oc	-	operating condition
s	-	shut off condition (no flow rate)
t	-	transient operating condition during valve shift or when piston reaches the end of the cylinder

# HYDRAULIC SYMBOLS

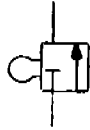
Relief Valve



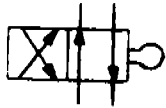
Check Valve



Pneumatic Actuated  
Two-Way Ball Valve



Pneumatic Actuated  
Four-Way Valve



Throttle Valve



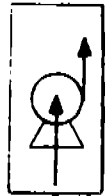
Shut-Off Valve



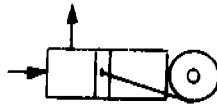
Pumps



High-Pressure  
Submersible Pump



Triplex Plunger Pump



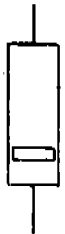
Flow Meter



Pressure Gage



Piston Accumulators



Bladder Accumulator



Solenoid Valve



Flow Switch



## SUMMARY

The flow work exchanger is a piston-and-cylinder arrangement whereby high-pressure brine discharging from the reverse osmosis desalination process is used to pressurize a portion of the feed stream to the process. Two cylinders are used in the device to achieve a nearly-steady output despite cyclical operation of each individual cylinder. Two low-pressure-rise pumps are required to make up for friction losses, one in the high-pressure brine and the other in the feed stream. Overall efficiency of the device exceeds 92% in the 9-to-18 gpm range at system pressures above 1000 psig.

Prototype development was reported by Cheng and Fan in O.S.W. Report No. 357, August 1968. Their program resulted in the construction of two work exchangers, one employing piston-type accumulators and the other employing bladder-type accumulators as the pressure exchange cylinders.

The objectives of the program described in this report were as follows:

- (1) Measure the performance of the piston and the bladder-type flow work exchange units.
- (2) Modify the two units to improve their start-up and operating characteristics and to improve the instrumentation.
- (3) Design an automatic control for synchronizing the piston or bladder movement to insure continuous output of high-pressure feed flow.
- (4) Complete a cost-effective analysis for the two types of work exchanger units to determine the engineering and economic limitations for scale-up of the work exchanger concept.
- (5) Select one of the two units for extended operation with an automatic control system.

- (6) Build the automatic control system and test it on one of the work exchangers to demonstrate satisfactory operation of the automatically controlled work exchanger.
- (7) Obtain component reliability information.

The piston and bladder type flow work exchangers were tested with fresh water at pressures up to 1500 psi and flow rates up to 20 gallons per minute. The efficiencies previously reported were verified by Dynatech's testing. Modifications were installed to improve unit operation and performance measurement. The modifications included improved instrumentation, pulsation-damping accumulators, corrosion-resistant check valves, and improved air bleeding systems.

The cost effective analysis revealed that work exchanger units can be built in sizes up to 600,000 gpd of fresh water output in a reverse osmosis plant with a 35% recovery factor. Larger capacities can be achieved by using units in parallel. Efficiencies of 90% to 95% are readily attained if losses in piping between the work exchanger and the membrane array are small. This report includes charts showing the efficiency and capital cost of work exchangers as a function of flow capacity and pressure level.

An automatic control was designed and constructed to automatically regulate the work exchanger operation so that an uninterrupted flow of high-pressure saline water will be delivered by the work exchanger despite variations in the flow rate of high-pressure brine discharged from the reverse osmosis process. The control system employs flow switches to detect the end of the stroke in the high-pressure side of the work exchanger cylinders. The controls then act to switch the brine inflow to the other side of the work exchanger in order to continue the work exchange process without interruption. The control system is made up of standard components.

The automatic control and piston type work exchanger components were operated satisfactorily for 619 hours (60,000 cycles) on fresh water at a pressure of 800 psig and flow rate of 8 gpm. The automatic control operated as expected to provide the proper valve switching functions. The accumulator walls and piston seals are the most critical components as shown by the amount of wear and deterioration experienced in the test program. The check valves, 4-way valve and system pumps operated without malfunctions.

## INTRODUCTION

In a study completed in 1968 for the Office of Saline Water (Contract No. 14-01-0001-1166), Professors C. Y. Cheng and L. T. Fan of Kansas State University demonstrated the feasibility of their flow work exchanger concept for use in the reverse osmosis desalination program. The flow work exchanger concept employs two displacement vessels as shown in Figure B-1 of Appendix B to simultaneously pressurize the feed stream while recovering energy from the high-pressure brine discharged by the process. The pressurized feed is pushed into the reverse osmosis system by the high-pressure brine in one displacement vessel at the same time that the depressurized brine is pushed out of the second displacement vessel by the low pressure feed stream. Proper synchronizing of the two vessels allows essentially constant flow of feed water and rejected brine through the machine.

The flow work exchanger itself neither requires nor delivers shaft work; energy is transferred directly from one fluid stream to another through a membrane or free piston. However, the work exchanger in its present form requires two auxiliary pumps which otherwise would not be needed in the reverse osmosis system. One pump boosts the pressure of the feed stream by the small amount needed to push the depressurized brine out of a vessel against atmospheric pressure. The second pump boosts the pressure of the reject brine by the amount needed to enable it to push the feed stream into the membrane array. This pressure rise must overcome brine-side pressure losses in the membrane system as well as smaller valve and pipe friction losses.

The first pump is a very conventional one; it operates near ambient pressure levels and is required to produce only a small head rise. The second pump also has to produce only a small head rise, but it must operate at a high average pressure level, e.g., 1500 psi. Submersible pumps, canned pumps, or magnetic drive pumps are required for this application. The submersible pump and its pressure enclosure was responsible for about 1/4 of the cost of the Kansas State test units.

Two different flow work exchangers were constructed and operated at Kansas State. Both units operated at the 1500 psi pressure level. The smaller unit delivered 9 gpm and used floating-piston displacement vessels. The larger unit delivered 18 gpm and used bladder-type displacement vessels. The construction of both units has been described in O.S.W. Research and Development Progress Report No. 357 published in August, 1968.

At the conclusion of this report, Professors Cheng and Fan made a number of suggestions for improving the construction of future units. These suggestions are summarized below:

- For the floating-piston type vessels, the provisions must be added to allow venting of air trapped beneath the piston when the machine is being started up.
- For the bladder-type vessels, bladders equal in capacity to the vessel volume must be provided in order to avoid pressure transients caused by elastic stretching of the bladder.
- Improved check valves with lower pressure drop and increased corrosion resistance should be provided.
- The high pressure booster pump must develop extra head to compensate for brine-side pressure drop in the membrane system. The present test units do not encounter such a pressure drop.
- An accumulator must be added to the high pressure lines to reduce water hammer accompanying valve actuation.
- Automatic controls must be added to synchronize the motion of the pistons or bladders in the two displacement vessels. Otherwise, the pressurized feed flow will not be steady.

The two prototype work exchanger units were shipped to Dynatech for implementation of these recommendations and for a period of reliability testing. Dynatech's program of work with these units was divided into two phases as follows:

Phase 1: Evaluation of performance and design of automatic controls.

Phase 2: Procurement, installation, and test of automatic controls.

This final report presents the results of both phases. The next section lists the specific objectives of this program.

## PROGRAM OBJECTIVES AND TASKS

The objective of the first phase of this program was to test, evaluate, and compare two different types of flow work exchanger devices for efficient recovery of the energy in the high pressure brine discharging from a reverse osmosis desalination plant. The two types of devices use as pressure vessels either two piston type accumulators or two bladder type accumulators. The specific tasks accomplished in the first phase of this project were the following:

1. Install, check out, and test the two existing flow work exchanger laboratory models built by Kansas State University for OSW.
2. Modifications were made as necessary to improve the system operation and to improve the instrumentation used to measure work exchanger performance.
3. The two modified flow work exchanger test units were retested to obtain performance measurements and to reveal potential operating problems.
4. Various alternative techniques for automatic control of the flow work exchangers were analyzed and compared to select the best control system for recommendation to OSW.
5. A cost-effective analysis was carried out to compare the two types of flow work exchanger systems and to determine which type of system was the best for use in a reverse osmosis process. The cost-effective analysis considered capital cost of equipment, its efficiency, its reliability of operation, and its scale-up potential.

The objective of the second phase of this program was to construct an automatic control system and operate one of the work exchangers with the automatic control to demonstrate control performance and system reliability. The specific tasks accomplished in the second phase of this project were the following:

1. The selected automatic control system was purchased and installed on the piston type flow work exchanger system.
2. The work exchanger was operated for 619 hours to demonstrate performance of the control system and to establish that the unit is operable with the control system designed for it.

## FLOW WORK EXCHANGER PERFORMANCE EQUATIONS

Derivations of the performance equations for the flow work exchanger are contained in Appendix A. There, the various elements of loss are listed and discussed and a comparison is made to the performance equations developed in Reference 1.

From Appendix A, the efficiency of the flow work exchanger is calculated from Equation (A-2) given below

$$\eta = 1 - \frac{W_l + W_h + W_c + W_a}{W_i} \quad (3-1)$$

For calculation of flow work exchanger efficiency from test data, the following assumptions are made:

1. Measured values of flow rate are used. This procedure automatically includes the effect of leakage losses in the laboratory system. (See Section A. 3.)
2. Measured values of pressure drop are used together with estimated pump and motor efficiencies to calculate the low and high pressure pump input power values. Since unnecessarily large control throttling losses are incorporated into the laboratory models, direct measurement of pump input power gives values that are significantly larger than those required in actual plant systems.
3. The power required by the pumps during transient operation is assumed to be equal to the pump power input at the normal operating condition. In reality, the transient input power may be slightly smaller.

With these assumptions, the equations to be used for calculation of the power terms in Equation (3-1) are the following:

$$W_i = (P_h - P_\ell) (Q_h) (0.321) \quad (3-2)$$

$$W_\ell = \frac{\Delta P_\ell Q_\ell}{\eta_{p_\ell} \eta_m} (0.321) \quad (3-3)$$

$$W_h = \frac{(\Delta P_h) Q_h}{\eta_{p_h} \eta_m} (0.321) \quad (3-4)$$

The term  $W_c$  represents the power input to the system controls. For the test units, power is needed to run the electric timer and the air solenoid

$$W_c = \text{Timer Power} + \frac{\text{Solenoid Power}}{2} \quad (3-5)$$

The term  $W_a$  represents the power input to the plant air supply to provide high pressure air for pneumatic actuation of control valves. This power input is calculated for the test units from the following equation:

$$W_a = \frac{m \Delta h}{\eta_c \eta_m} (778) \quad (3-6)$$

The air is supplied from a receiver tank which is periodically recharged by a compressor. The efficiency values used should be typical values achieved by small plant air supplies.

## TEST PROGRAM ON TWO PROTOTYPE FLOW WORK EXCHANGERS

4.1 Work Exchanger Configurations and Modifications

Two basic configurations of the flow work exchanger system were tested in several forms in this test program. The two systems were similar except for the type of energy exchange pressure vessel used: bladder type or piston type accumulators. A complete description of the two original configurations built by Kansas State University is presented in Reference 1. This reference includes pictures, system schematics, and equipment specifications. The original configuration for each type of work exchanger tested in the present program is the same configuration and equipment pictured and described in Reference 1.

Figure 4-1 shows a system schematic of the original configuration for both types of units. The flow rate of the low pressure feed flow ( $F_1$ ) and the high pressure flow ( $F_2$ ) are measured by magnetic follower rotameters. The pressures are measured in five locations using Bourdon tube type gauges. In the original system the pressure loss on the low pressure side of the system is determined from the difference between the readings on gauges  $P_1$  and  $P_2$  after correction for the difference in elevation. No measurement is made of the pressure loss on the high pressure side of the system. The piston accumulators used are standard units without modification. The bladder accumulators used as pressure exchange vessels have been modified to enlarge the port attached to the bladder. The hole size in the stem of a standard transfer barrier bladder assembly is approximately 0.38 inches in diameter. The bladder assembly was modified by Kansas State University by cutting off the standard stem, enlarging the hole to 0.75 inches and attaching a new flanged stem with screws threaded into the metal piece molded into the standard bladder. The hole in the pressure vessel was enlarged to accommodate the enlarged stem outer diameter.

After completion of a series of tests on the original configurations of both the bladder type and the piston type work exchangers, both systems were modified

and retested. The modifications added to the two units are shown on Figure 4-2. In addition to the modifications shown on Figure 4-2, an air purging system was added to both systems to allow purging of air from below the pistons and bladders. The piston purge system consisted of a poppet relief valve mounted in the piston as shown on Figure 4-3. The relief valve was set to open when subjected to the shut-off head of the low pressure pump. It is open during startup of the work exchanger or when the piston is resting against the end of the accumulator, since there is only a very small pressure difference across the piston during normal operation.

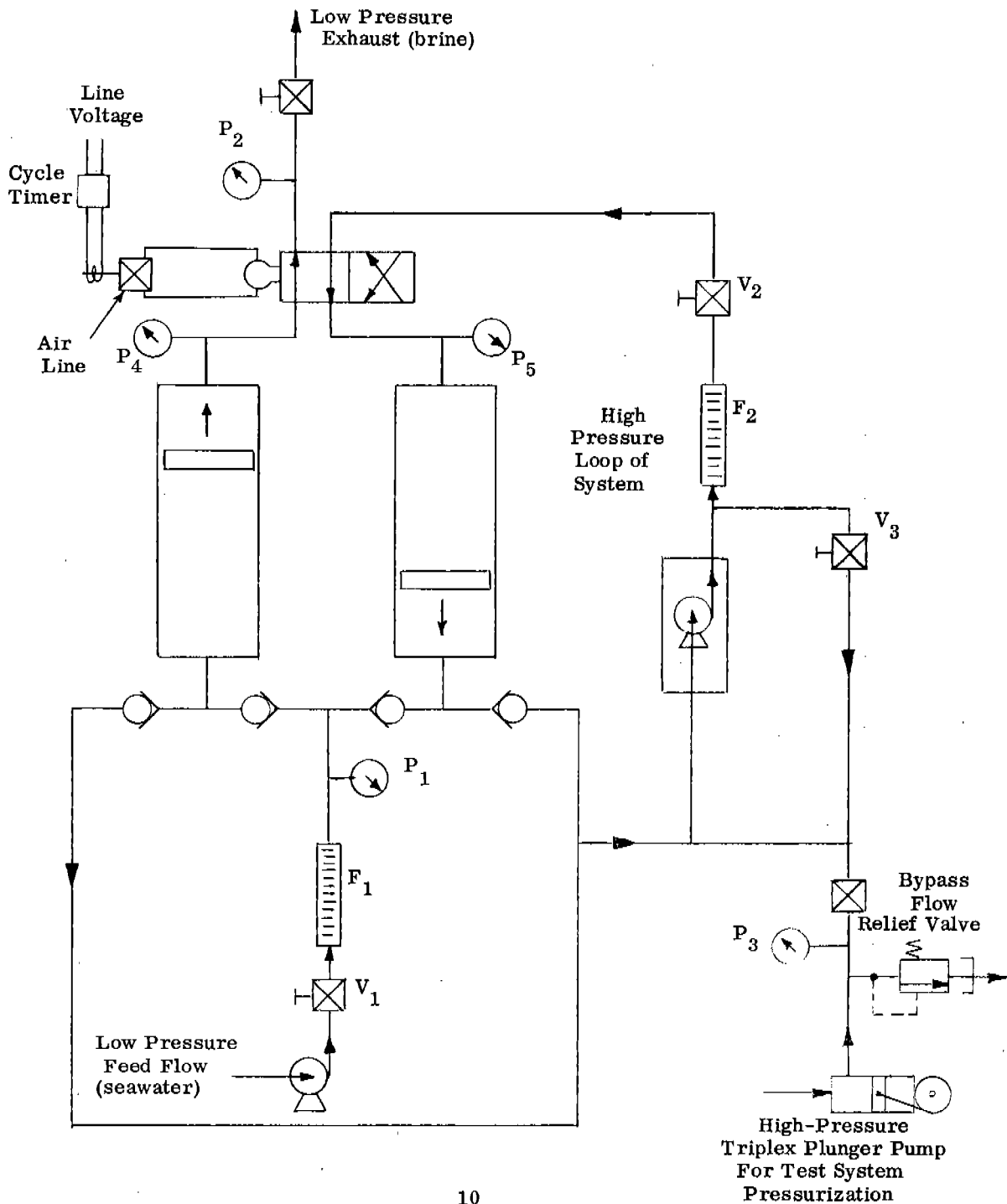
The bladder purge system is also shown on Figure 4-3. To purge air during startup, the bladder is first expanded from above to force most of the air out of the pressure vessel and close the valve. Water is then admitted from below and the trapped air is bled out through the purge valve. The bladder system is more difficult to purge of air.

Figure 4-3 shows the typical cross-section of a modified bladder accumulator, a standard piston accumulator, and the three piston configurations tested in this program.

The modifications of the prototype work exchangers were made for the following reasons:

1. The differential pressure gauges were added to improve the accuracy of the pressure loss measurement and to obtain a measurement of the pressure loss on the high pressure side of the system.
2. The original carbon steel check valves were replaced with brass check valves to prevent sticking problems.
3. The pulsation damping accumulators were installed to reduce the system pulsations caused by the triplex plunger pump and by pressure waves caused by the rapid closure of the 4-way control valve.
4. The high pressure gauge ( $P_G$ ) was installed to observe the steadiness of the high pressure discharge flow coming from the work exchanger.

Figure 4-1  
Original Flow Work Exchanger Test System Schematic  
(see nomenclature for standard symbols)



5. The air purge systems were installed to allow easier removal of air which is trapped below the pistons and between the bladder and shell of the pressure exchange accumulators.

The differential pressure gauges  $\Delta P_l$  and  $\Delta P_h$  shown on Figure 4-2 are 5000 psi gauges manufactured by the Mid-west Instrument Company. The scales are 0 - 25 psid with an accuracy of  $\pm 0.5$  psi. The pulsation damping accumulators are standard hydropneumatic bladder type accumulators manufactured by the Greer Olaer Products Company. The sizes are 1 gallon and 2-1-2 gallon capacities with the larger unit mounted at the pump. Two pictures of the modified piston type flow work exchange test unit are shown in Figures 4-4 and 4-5.

#### 4.2 Start-Up and Operation Procedures

The following start-up procedure was used during the test program.

1. When using the modified system, the hydropneumatic accumulators were charged to 600 psig.
2. The triplex plunger pump was bled of air by rotating the shaft by hand with the pump bleed ports open until all air bubbles stop.
3. The pneumatic air supply is turned on and regulated to 40 psig.
4. The air is bled from the lower part of the work exchanger system. In the original bladder system, the air was removed by loosening the nut on the bladder stem and then pumping water into the system with the low pressure pump. In the original piston system, a tube was inserted into the bottom of the piston accumulator until it touched the bottom of the piston. The low pressure pump or city water pressure was then used to fill the system and force the air out. In the modified systems, the devices shown on Figure 4-3 were used as described in Section 4.1.
5. The upper part of the system is then filled and pressurized. Air is bled out through valves, gage connections, and the exhaust line.

Figure 4-2

Modified Flow Work Exchanger Test System Schematic

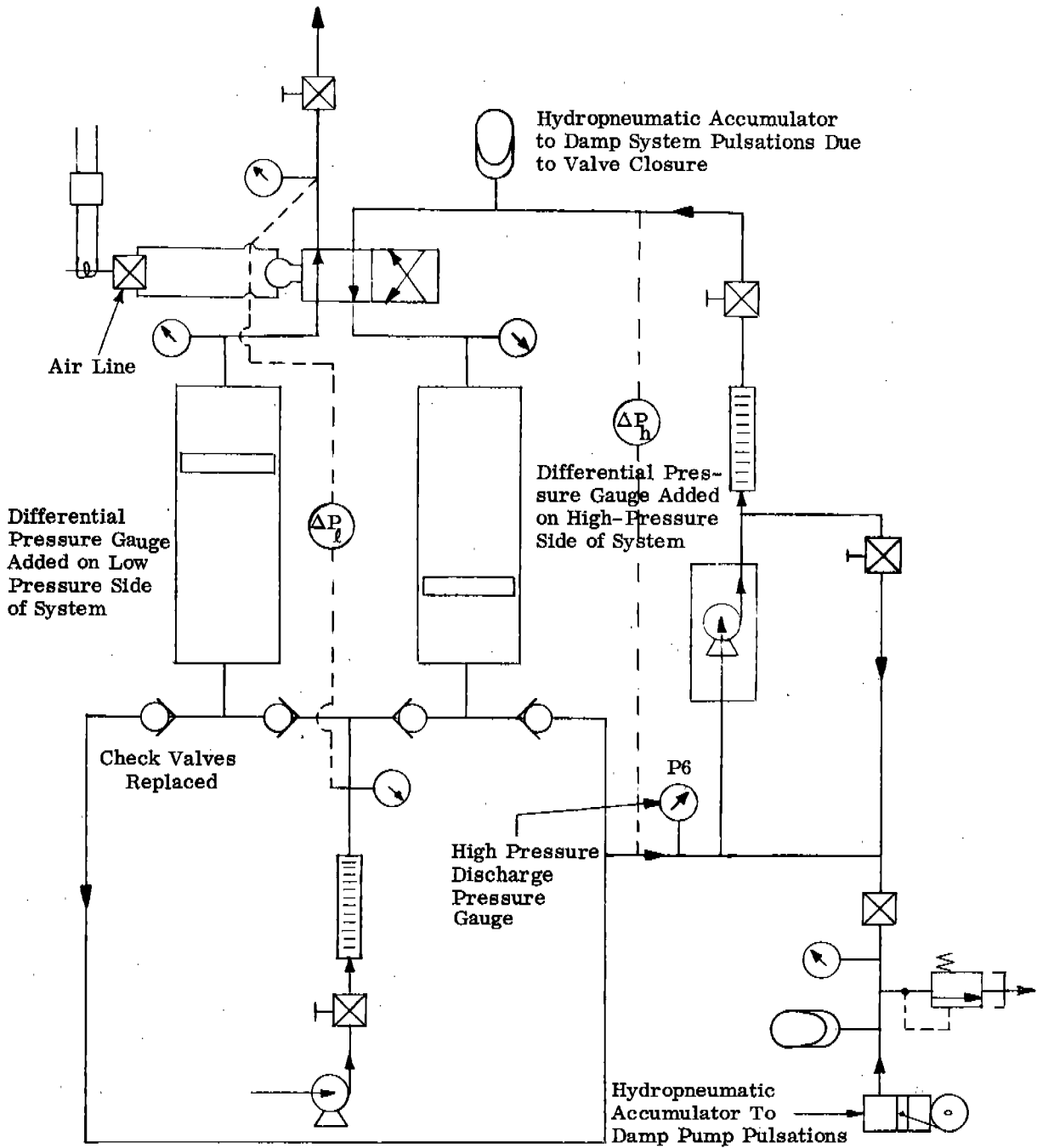
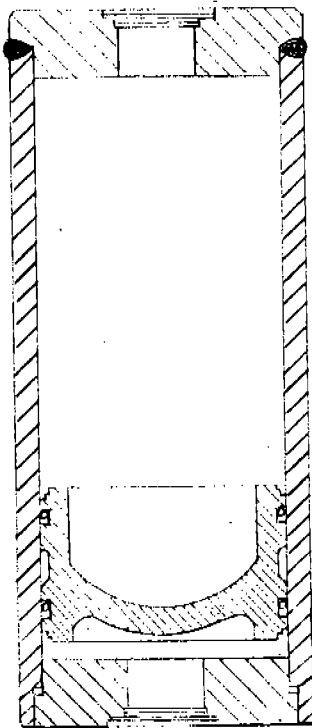
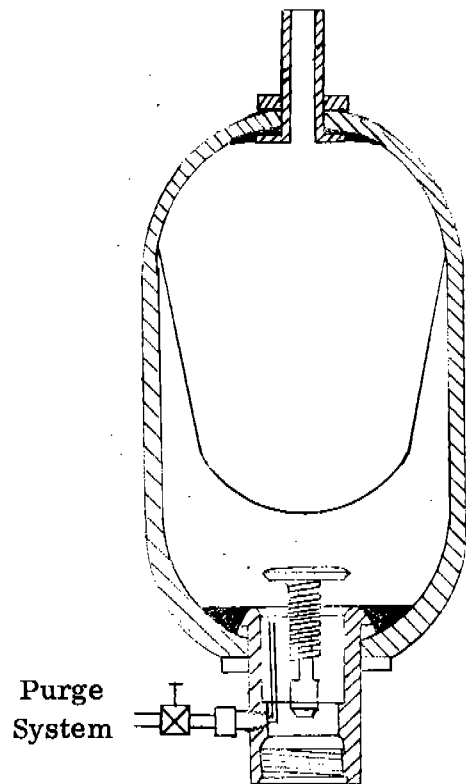


Figure 4-3

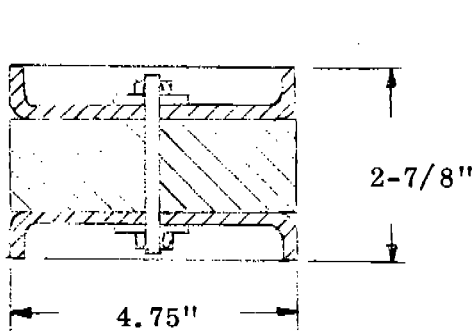
Accumulator and Piston Configurations



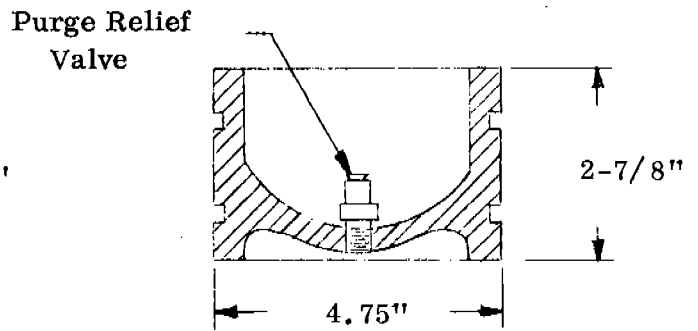
Piston Accumulator  
With Standard Piston



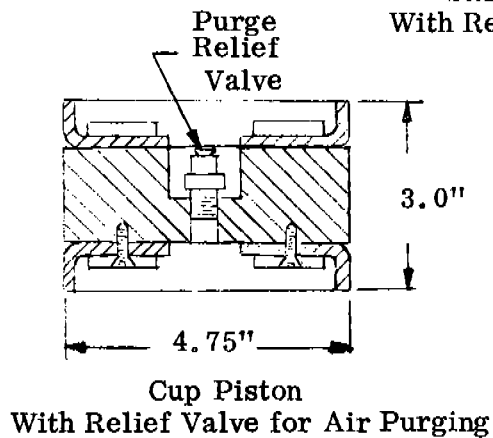
Bladder Accumulator



Solid Cup Piston



Standard Aluminum Piston  
With Relief Valve for Air Purging



Cup Piston  
With Relief Valve for Air Purging

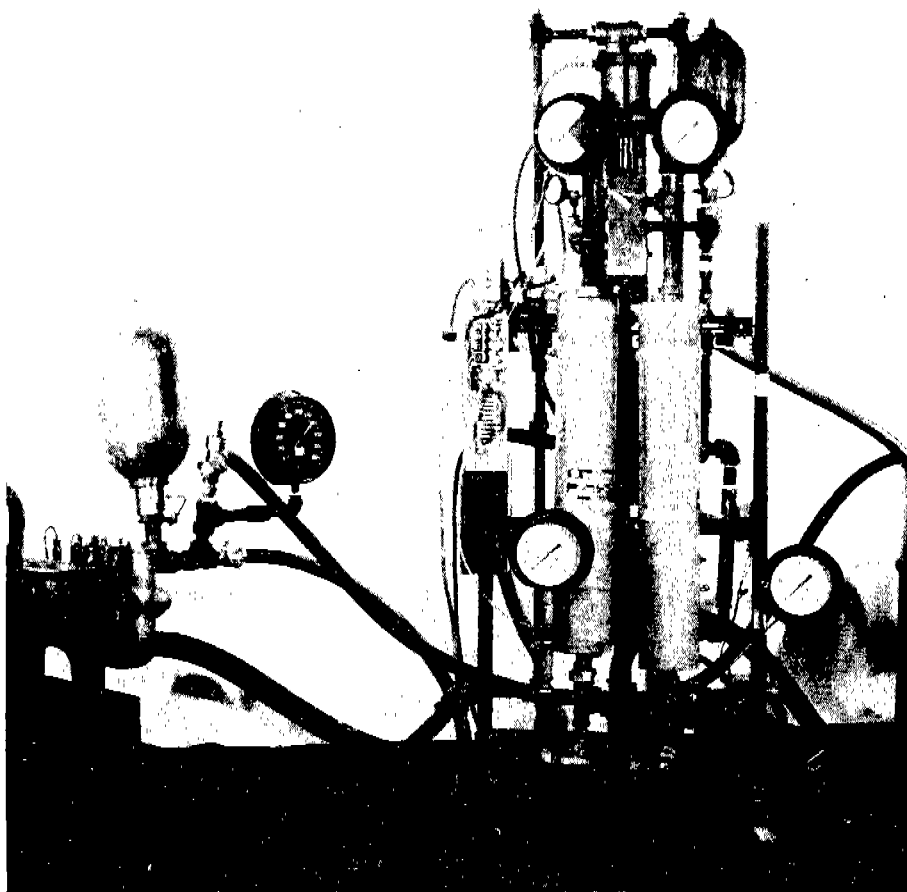


Figure 4-4  
Picture of Modified Piston Flow Work Exchanger  
Front View

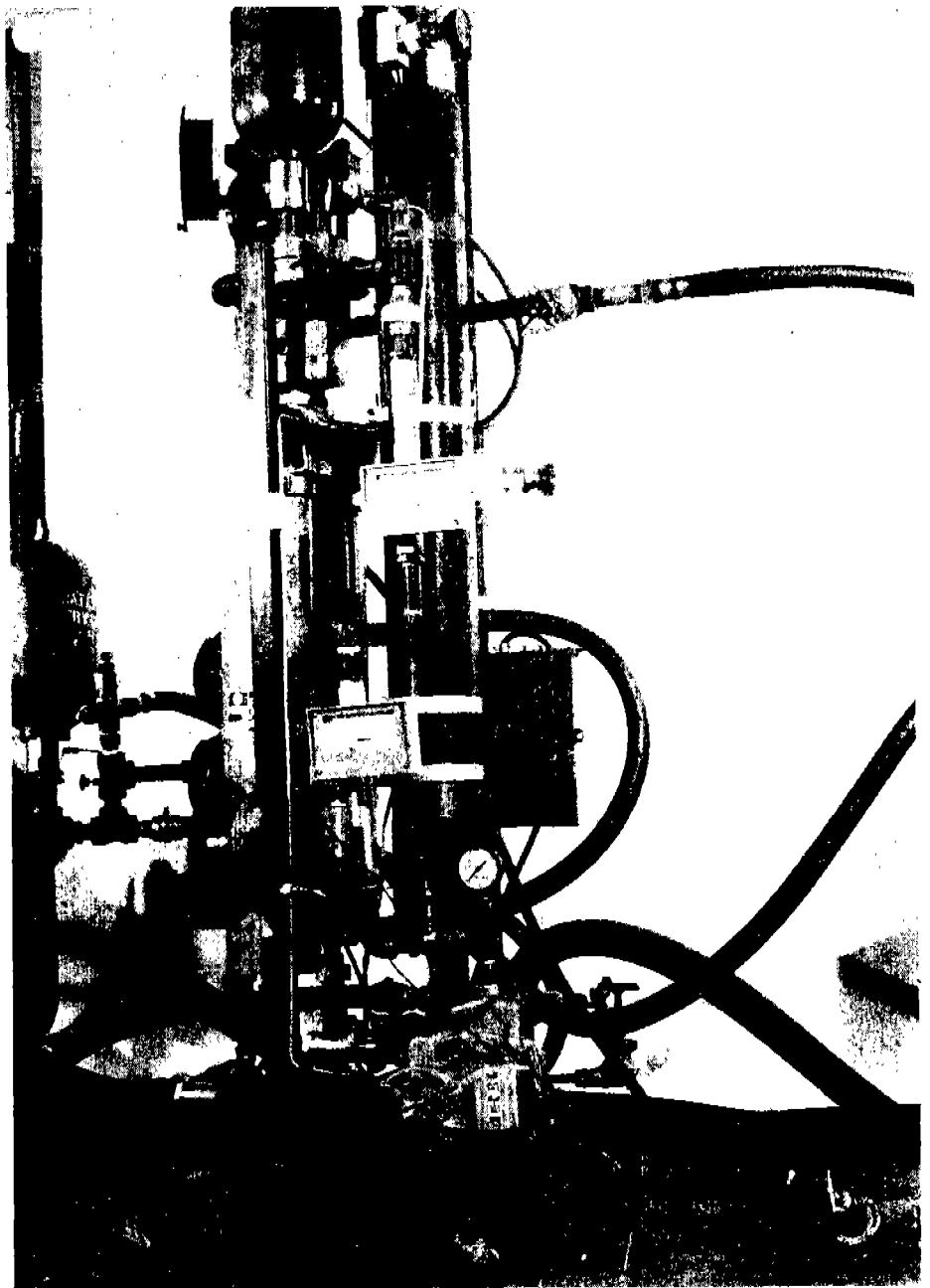


Figure 4-5  
Picture of Modified Piston Flow Work Exchanger  
Side View

6. Either the high pressure submersible pump or the low pressure pump is used to move both pistons to the same end of the accumulators.
7. The timer is positioned at the beginning of a cycle.
8. The high pressure triplex pump is turned on and set at low pressure (about 200 psig).
9. The low pressure pump, submersible pump, and timer are turned on simultaneously.
10. The flow rates of the low pressure and the high pressure water are regulated by valves  $V_1$ ,  $V_2$ , and  $V_3$  of Figure 4-1.

The operation of the flow work exchanger is different when operated by itself as shown in Figure 4-1 than when it is operating as part of a reverse osmosis system. When operating by itself, the triplex plunger pump is used only to pressurize the system up to pressures of 1500 psi. The flow entering the triplex pump is continually exhausted through the relief valve and back to the reservoir. Flow entering the low pressure pump flows into the bottom of one of the accumulators. When the 4-way valve switches, this fluid is pressurized by the high pressure side of the system and the flow is forced out of the accumulator through the check valve and into the suction line of the high pressure submersible pump. This pump increases the pressure of the fluid enough to make up for piping losses and the fluid flows into the top end of one of the accumulators. When the valve switches again this fluid is discharged through the exhaust line and back to the water reservoir. A description of the operation of a work exchanger with a reverse osmosis system is included in Section 1.

The most simple way to synchronize the piston or bladder motion to provide a continuous flow of high pressure fluid is to set the low pressure feed flow to a larger value of flow rate than the high pressure flow. This means that the piston moves faster on the exhaust stroke than on the high pressure stroke and therefore will eventually operate with the piston reaching the end of the cylinder each cycle during the exhaust stroke. When both cylinders are operating in this manner, the low pressure feed flow can be reduced to the same value as the high pressure flow

to minimize the time interval during which the piston is bottomed during the exhaust stroke. If the level of low pressure flow is slightly larger than the high pressure flow, the prototype work exchanger will operate for hours or even days at a time with uninterrupted high pressure flow. Some of the automatic control methods discussed in Section 5 use this type of operation. The bladder type flow work exchanger operates in the same manner.

#### 4.3 Test Program and Results

A series of tests were completed with both the bladder and piston work exchanger test units in their original and modified configurations. The tests were run at a range of flow rates and with system pressures up to 1500 psig. A summary of the test program is given on Table 4-1.

The data recorded during most of the tests includes the following information:

1. Differential pressures  $\Delta P_l$  and  $\Delta P_h$ .
2. Pressure gauge readings and pressure fluctuations on gauges  $P_1$  through  $P_6$ .
3. Flowmeter readings and fluctuations on rotameters  $F_1$  and  $F_2$ .
4. Cycle time.
5. Air supply pressure.
6. Motor current and voltages for the pumps during the test of modification 1 on the bladder unit.

The test data is presented in Figures 4-6 and 4-7 in the form of measured pressure loss and calculated work exchanger efficiency as a function of flow rate. The pressure losses and efficiencies are presented for one basic bladder unit and for the piston unit with both cup type pistons and also with standard O-ring pistons. The performance of the two original configurations was measured first with two gauges on the low pressure side of the system (see Figure 4-1) and then with differential pressure gauges on both the low pressure and high pressure sides of the system.

Table 4-1  
SUMMARY OF THE PROGRAM

System Configuration	Piston Unit		Bladder Unit	
	Data Taken	Length of Test	Data Taken	Length of Test
Original Configurations As Received From KSU	200, 600, 1000 psi 1500 psi - 3.0, 4.5, 6.0, 7.5, 9.0 gpm	2 hr. test 8 hr. observation	200, 700, 900 psi 10, 15, 20 gpm	2 hr. test 8 hr. observation
System Modification No. 1 (1) Original piston or bladder (2) Two accumulators added (3) $\Delta P_t$ , $\Delta P_h$ gages used (4) Added second high pressure gage (5) Air bleed for low-pressure side (6) Carbon steel check valves replaced by Brass Check Valves	<u>Original Piston - No Bleed</u> 200, 800, 1200 psi 4, 6, 8, 10 gpm  <u>New Cup Piston - With Bleed</u> 200, 800, 1000, 1200 psi 1500 psi - 4, 6, 8, 10 gpm	2 hr. test  7 hr. test	200, 800, 1000, 1200, 1500 psi 10, 15, 20 gpm	3-1/2 hr. test
System Modification No. 2 Piston Unit Only (Modification No. 1 and pistons as in next column).	<u>Standard - Unmodified Piston</u> 200, 800, 1000, 1200 psia at 4 gpm - 6, 8, 10, gpm at 200 psi <u>Standard - Relief Valve Piston</u> <u>Added</u> 200, 800, 1000, 1200, 1500 ps at 4, 6, 8, 10 gpm	4 hr. test  6-1/2 hr. test		

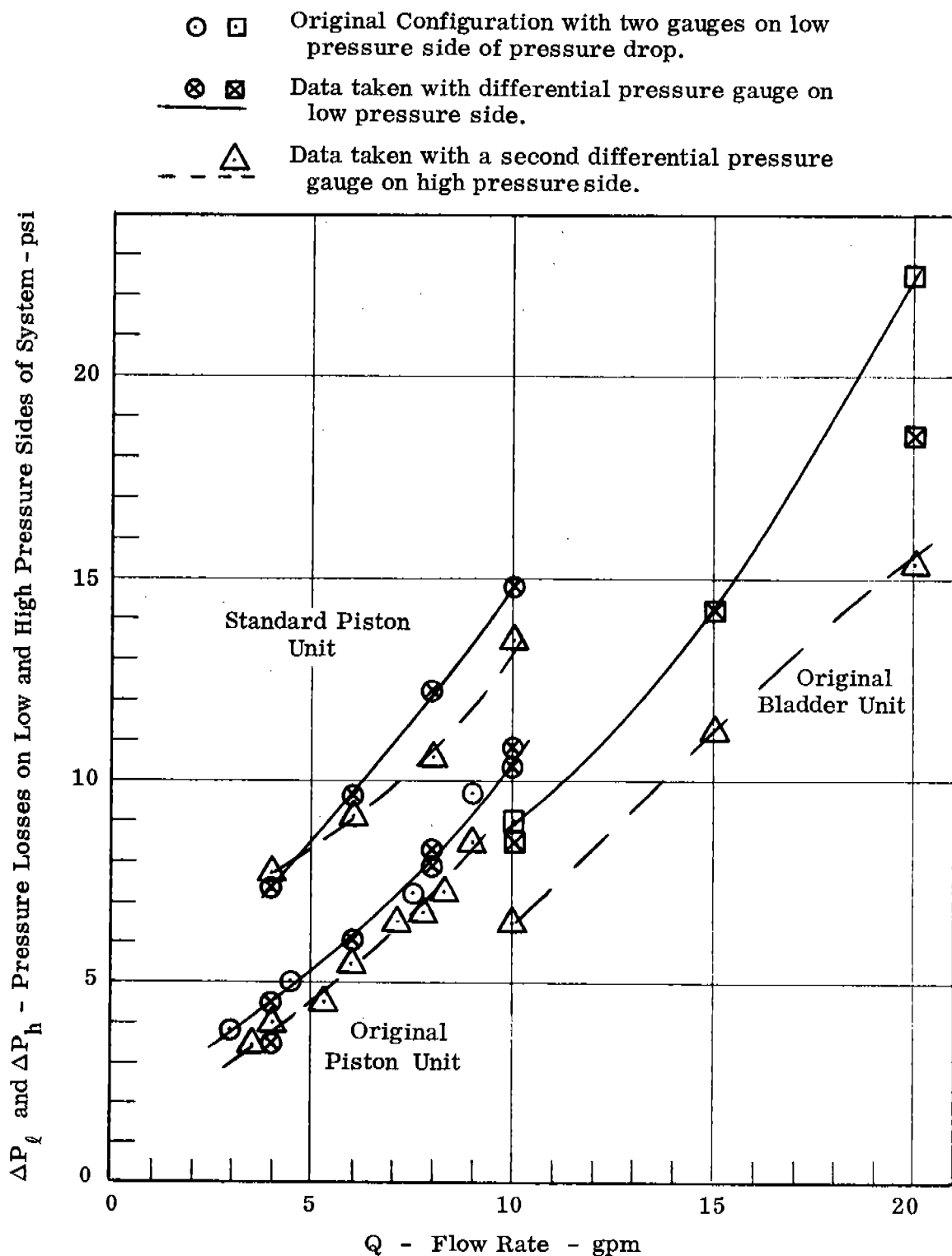


Figure 4-6. Measured Pressure Losses for Bladder and Piston Type Flow Work Exchangers

- 1500 psig system pressure
- ⊗— 1000 psig system pressure
- Open Symbol 2 separate gauges on low pressure side of system
- x Symbols Differential pressure gauges used

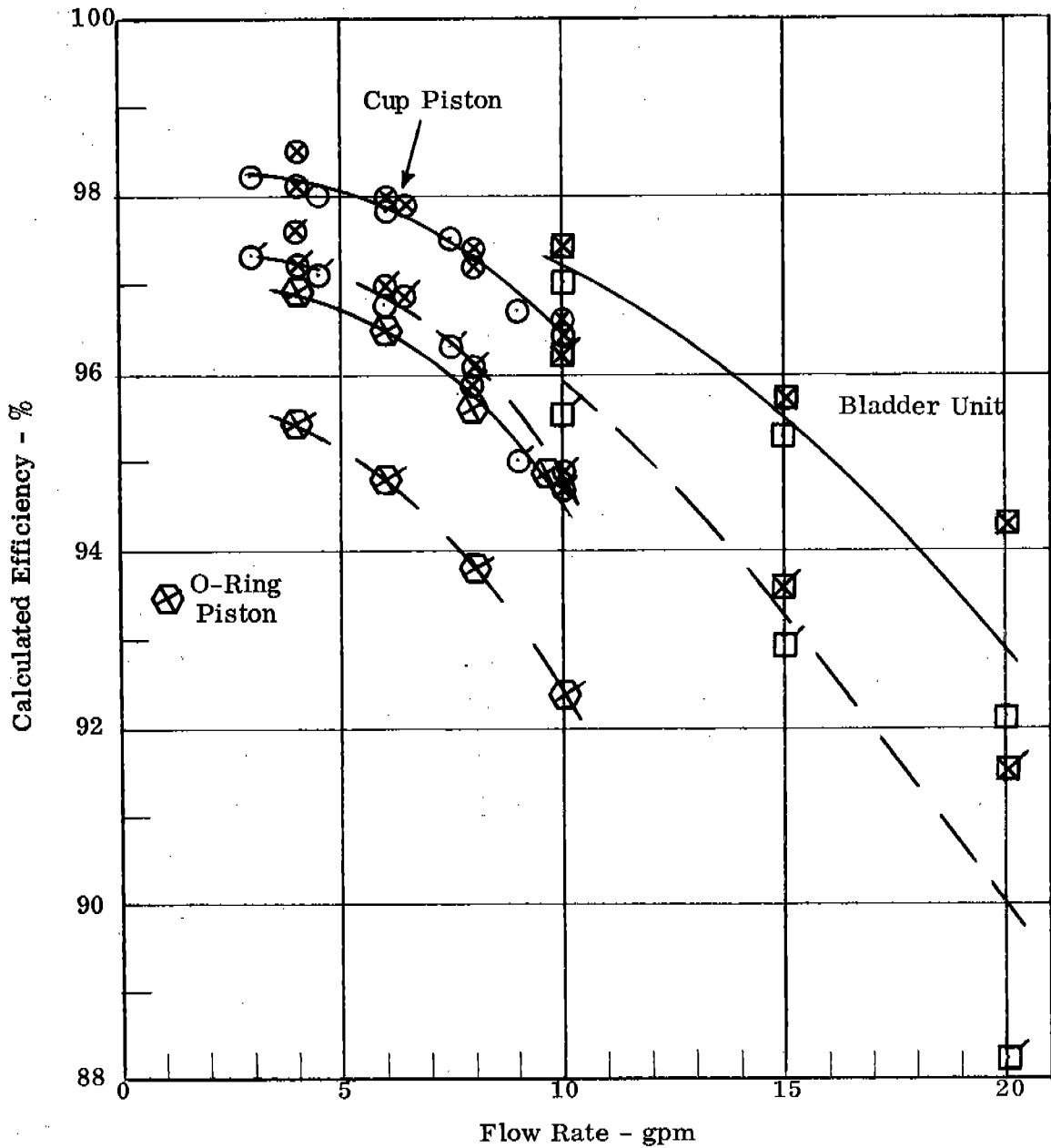


Figure 4-7. Calculated Work Exchanger Efficiencies Based Upon Measured Pressure Losses

The differential pressure readings on the low pressure side of the system agree well with the values measured by two separate gauges. The differential pressure loss on the high pressure side of the system is slightly smaller than the differential pressure loss on the low pressure side of the system due to piping differences. The standard O-ring piston has a pressure loss of about 3 psi larger than the cup type piston; but, as shown on Figure 4-7, this small increase in loss has an insignificant effect on work exchanger efficiency. The original and new cup type pistons shown on Figure 4-3 were both tested; data for both are included on Figure 4-6. The inner diameter of the piping on the two systems is approximately the same. This explains why the pressure loss versus flow rate is almost a continuous curve between the two systems.

The flow work exchanger efficiencies calculated from the pressure loss data are shown on Figure 4-7. These efficiencies are calculated by using Eq. (3-1) and the other relationships presented in Section 3. Since the pumps mounted on the test work exchanger systems are oversized in both head and flow capacity, the motor input power was not used to calculate efficiency. Instead, the power input to both pumps is calculated using the following procedure.

1. The pump specific speed is calculated and Figure B-9 is used to select an efficiency value representative of the efficiency of a properly sized pump to supply the measured head and flow rate.
2. A motor efficiency of 80% is combined with the pump efficiency in Eqs. (3-2) and (3-3) to calculate motor input power.
3. The control power input and equivalent air power input were calculated to be approximately 8.7 ft lb/sec for the piston unit and 10.4 ft lb/sec for the bladder unit. These values are small compared to the motor input powers.
4. Eq. (3-1) is used to calculate the flow work exchanger efficiency.

Since measured pressures are used for calculation of the efficiency, several very important pressure losses that would occur in reverse osmosis systems are

excluded. Inclusion of these losses would reduce the levels of efficiency below those shown on Figure 4-6. These losses include the following:

1. Pressure losses in the external high pressure piping connecting the flow work exchanger to the reverse osmosis plant. These losses could be substantial.
2. Pressure losses in the low pressure feed and exhaust lines.
3. Throttle valve losses for oversized pumps or automatic control system valves.
4. Pressure losses on the brine side of the reverse osmosis membrane system.

Only the first three losses should be included in work exchanger efficiency although the submersible pump must be sized to make up all four of these losses. Section 6.2.3 includes a discussion of the effect of increased pressure drop on work exchanger efficiency.

#### 4.4 Test Observations

Observations of the system operating characteristics were made in the following five areas:

1. Reaction of system to air purging methods.
2. System flow and pressure pulsations.
3. Reliability of system components.
4. Control valve switching time.
5. Bladder stretching pressures.

The system operation is quite sensitive to air trapped below the piston or bladder. If air is trapped there, the pressure gauges on the high pressure side of the system will drop drastically at the beginning of a work exchange cycle and slowly build back up to the system operating pressure. This occurs because the trapped air pocket must be compressed in size by the introduction of liquid into

the system. When no air is trapped in the system, the high pressure gauges ( $P_4$  and  $P_5$ ) jump immediately to the system pressure when the valve switches. The purging of air in the original work exchanger configurations was considerably more difficult than with the modifications shown on Figure 4-3. If no attempt was made to purge the units, and the units were run for an hour, no improvement in performance was found to occur. This showed that an improved purging arrangement was essential.

The modified piston unit with the piston relief valves can be started up and purged in a matter of one or two minutes after the high pressure triplex pump has been purged. The bladder unit purging takes longer and may or may not be as complete as in the piston unit.

When the 4-way control valve switches, fluctuations occur on both the flow meters and the pressure gauges. A summary of these observations is given on Table 4-2. The flow meter oscillations are a function of the actual flow variation during the valve shift and also the looseness of the pointer bearings. The  $F_1$  flowmeter indications appear to be unaffected by the presence of the accumulators. The switching fluctuations of the  $F_1$  flow meter for the piston unit seems to be larger than for the bladder unit but this may be related to which meter is used. The presence of the accumulators reduces the high pressure  $F_2$  flow meter switching fluctuations significantly. In fact, when this flow meter was received from Kansas State University, the meter was damaged severely as though it had experienced a strong flow reversal. In some of our testing without accumulators, the float of this meter could be heard striking the stop during the valve switching.

The pump discharge pressure oscillations were not reduced when an accumulator was located at the pump. If anything, the oscillations become worse. The high pressure gauge located at the high pressure feed flow outlet ( $P_6$  on Figure 4-2) showed reduced oscillations for the piston unit but not for the bladder unit. However, a Bourdon tube gauge gives only an approximate measure of the actual pressure variation.

Table 4-2

**SUMMARY OF SYSTEM FLOW AND PRESSURE FLUCTUATIONS  
DURING VALVE SWITCHING**

Measurement	Piston Unit		Bladder Unit	
	No Accumulators	With Accumulators	No Accumulators	With Accumulators
Flowmeter $F_1$	oscillates $\pm 30\%$	oscillates $\pm 30\%$	oscillates $\pm 3\%$	not recorded
Flowmeter $F_2$	drops to zero, then to 100%, then oscil- lates	oscillates $\pm 20\%$	drops to zero, then to almost 80%, then oscillates	not recorded
Pressure $P_3$	$\pm 100$ psi at all levels	$\pm 100$ psi at all levels above 600 psig	$\pm 50$ psi at all levels	$\pm 100$ psi at all levels above 600 psi
Pressure $P_6$	drops 60 psi at 200 psig	drops 30 psi at 1000 psig, 40 psi at 1500 psig	drops 60 psi at 200 psig	drops 70 psi at 1000 psig, 100 psi at 1500 psig

The following conclusions were reached on the use of pulsation damping accumulators in the flow work exchanger:

1. The accumulator located near the high-pressure pump does not appear to reduce pressure pulsations produced by the pump.
2. The accumulator located upstream of the valve significantly reduces the flow meter fluctuations and slightly reduces the pressure pulsations.

Several observations were made regarding component reliability during the experimental program. These observations were:

1. The carbon steel check valves on the original unit stuck on several occasions. These valves were replaced with brass check valves and no further problems were encountered.
2. The modified bladder assembly of the standard bladder type accumulator used in the test unit was very unreliable. Leaks developed around the stem modification; these leaks could be eliminated by a better design. Of greater consequence is the fact that two different bladders developed a series of small holes around the top of the bladder. One of the bladders received from Kansas State University was like this and the new bladder purchased to replace it also developed holes at some time during the test program. The holes appear to be punctures from the inside out and are apparently caused by the bladder sticking to the phenolic coated shell of the accumulator vessel. Since our system is a closed loop and recycles the same water continuously, the temperature of this water will reach a value of about 130° F after an hour of operation. If the system is then shut down the bladder may stick to the shell. When started up the next time, some of the points that were stuck may pull and form the puncture type holes. The water temperature of an actual work exchanger system in a reverse osmosis process would never be significantly higher than ambient. However, this test problem points out the delicate nature of the bladders.
3. The piston accumulators may also have problems due to the phenolic coating. When the piston accumulators were received from Kansas

State University the inner surface of one of the accumulators was corroded and rough in a ring-shaped region about 1 inch long. Water apparently had been left in part of the unit and the piston or free surface rusted at this point. Some manufacturers provide chrome plating on their shells. This plating may be better for seawater service.

Valve switching times were measured for the four-way valve on the piston unit. The stopwatch values were about 0.25 seconds for valve movement from one end to the other. The transient operating period shown on the gauges appears to be slightly less than this value.

A measurement of the pressure required to stretch one of the accumulator bladders to the full vessel size gave a value less than 6 psi. Since this value is small, a new larger bladder was not obtained to eliminate the bladder stretching losses.

#### 4.5 Test Program Conclusions

1. The measured flow work exchanger pressure losses are low and the calculated efficiencies are in the range of 92% to 97% for a 1500 psi system. However, line losses in the pipes and fittings needed to install the flow work exchanger in a reverse osmosis plant may be significant depending on the pipe size and flow rates. These losses can reduce the flow work exchanger efficiency significantly.
2. Piston accumulators are more reliable and less susceptible to failure than bladder accumulators under the service conditions encountered by a flow work exchanger.
3. The piston test system was operated successfully without attendance for an hour with the low pressure flow slightly larger than the high pressure flow. The high pressure flow was maintained at a steady value. This manner of operation is similar to that recommended for operation with a simple automatic control system.

4. The use of pulsation damping accumulators will reduce the high pressure flow rate variations and pressure fluctuations during valve switching transients to the following levels:

flow rate  $\pm 20\%$

pressure  $\pm 40$  psi at 1500 psig for piston unit

5. The bladder assembly of a bladder type accumulator is very difficult to modify successfully. This would be particularly true for larger flow rate units than were tested in this program.

6. The piston unit is easier to purge of air during startup than a bladder type unit.

DESIGN OF CONTROLS FOR AUTOMATIC OPERATION OF A  
WORK EXCHANGER5.1 Background

The operation of the OSW work exchangers was originally controlled by an electrical timer which cycled the 4-way valve to alternate the high-pressure brine inflow to the two exchanger vessels. In order to maintain continuous flow, the stroke of the two pistons or bladders had to be made identical (insofar as possible) by the operator who adjusted valves controlling the flow rates of the high-pressure membrane discharge brine flow and the low pressure saline water feed stream as they enter the work exchanger. The operator had to check the operation of the unit frequently to assure that the pistons or bladders were not bottoming during the cycle, thereby reducing the flow rate of the unit and causing discontinuous delivery. It was apparent that an automatic flow synchronizing control system would be essential if flow work exchangers were to be practical in reverse osmosis systems.

5.2 Objectives for the Automatic Control System

The automatic flow synchronizing control system should meet three objectives:

1. Steady Output Flow: The high-pressure saline water output flow from the work exchanger to the reverse osmosis membranes should be as steady as possible. Switching transients should be minimized.
2. Minimum Losses: The additional pressure losses and inefficiencies caused by the automatic control system should be small.
3. Automatic Startup: The control system should be compatible with automatic startup of the work exchanger without attention from an operator.

In addition, the automatic control system should be made up of standard components, be of reasonable cost, and be reliable for prolonged service.

### 5.3 Control System Design Considerations

The work exchanger flow arrangement is shown schematically in Figure 5-1. Several considerations which hold for any suitable control system can be immediately identified:

- The volumetric flow rate of the unit is set by the high-pressure brine input feed stream.
- The control valve switching must be regulated by some type of sensor in the high-pressure portion of the system since the high-pressure saline water delivery to the membranes is to be continuous.
- The low-pressure flow rate must be regulated in such a way that the piston or bladder in the low-pressure vessel reaches the end of its stroke not later than the one in the high-pressure vessel.
- There are two principal tasks for the control system to perform:
  - Switch the brine-side flows in response to an appropriate signal
  - Regulate the piston or bladder motion in the low-pressure side of the system so that the end of the stroke is reached at or before the time when switching occurs.

The signal to cause switching of the brine-side flows can be obtained from one of several possible sources:

- Sense piston or bladder position at the end of the stroke in the high-pressure vessel.
- Sense the volume of flow and actuate the switch when the full displacement capacity of the high-pressure vessel is approached.
- Sense the end of delivery of flow from the high-pressure vessel which occurs when the stroke limit is reached.

The regulation of the piston or bladder motion in the low-pressure side of the work exchanger system can be accomplished by manually setting the throttle

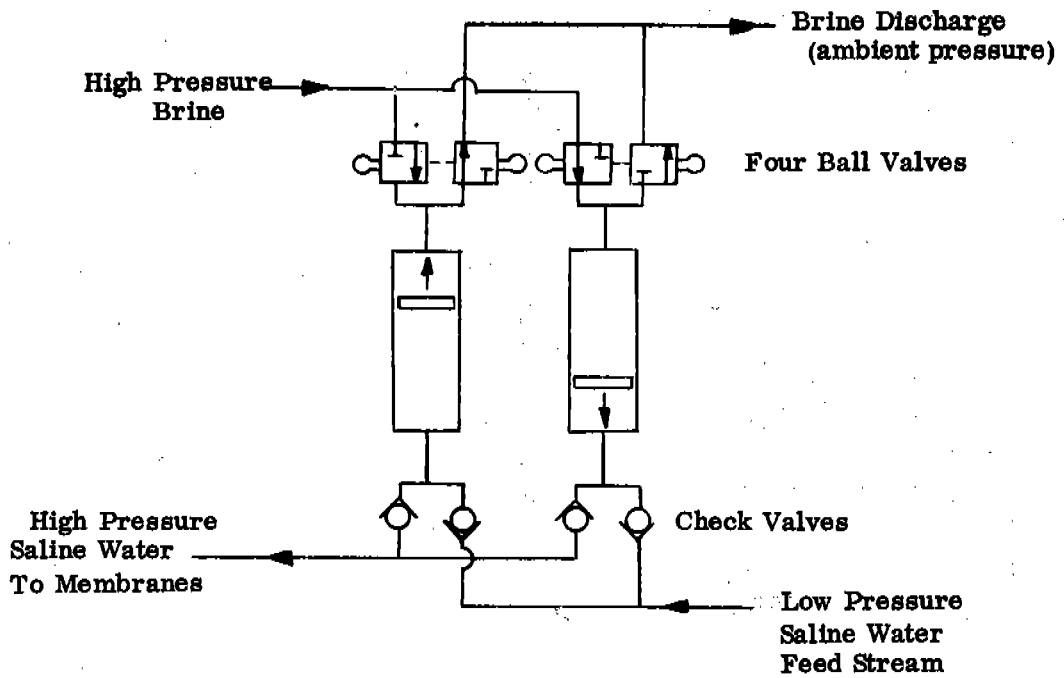


Figure 5-1: Schematic Arrangement of the Work Exchanger

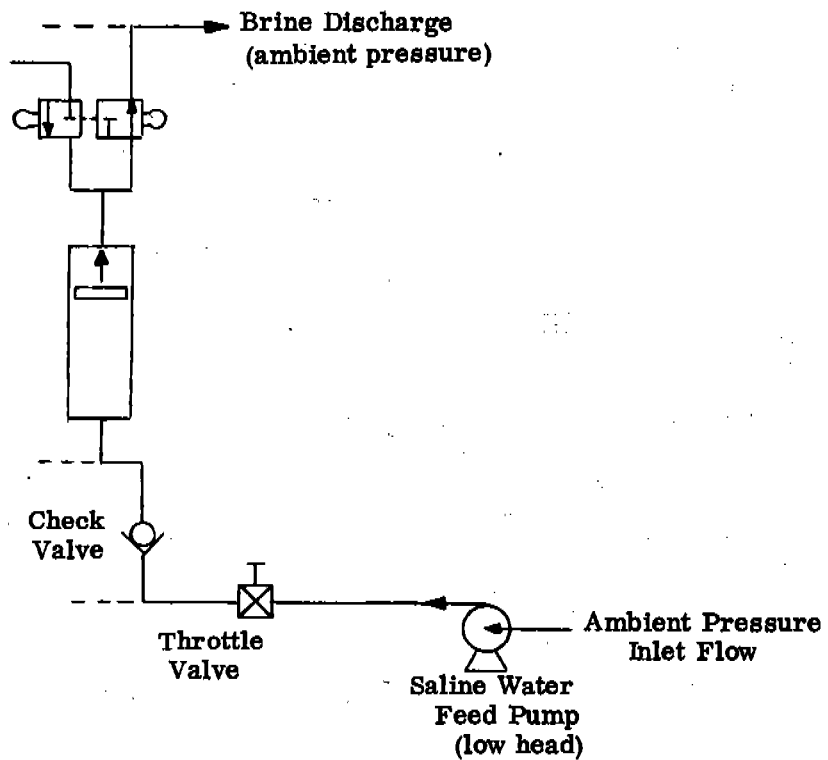


Figure 5-2: Low-Pressure Side of the Work Exchanger Flow Path

valve shown in Figure 5-2, so that the feed pump always delivers a flow rate slightly larger than the maximum brine-side flow rate. This will assure that the low-pressure piston will always reach the end of its stroke faster than the high-pressure piston. If the brine flow rate drops off due to changes in the operation of the reverse osmosis process, the time interval for which the low-pressure piston is bottomed will increase. The feed pump will operate for a fixed time interval at constant flow followed by a variable time interval at zero (stalled) flow. This type of operation is shown on the pump characteristic curve of Figure 5-3.

It is expected that reverse osmosis desalination plants will be operated at constant flow rates for long periods of time. Variations in demand for the fresh water product will be accommodated by storage systems such as water towers or ponds. Therefore, the feed pump can be sized to operate efficiently at a flow rate only slightly larger than the work exchanger design flow rate. The throttle valve will be only slightly closed so that the associated losses in the low-pressure feed system will be low.

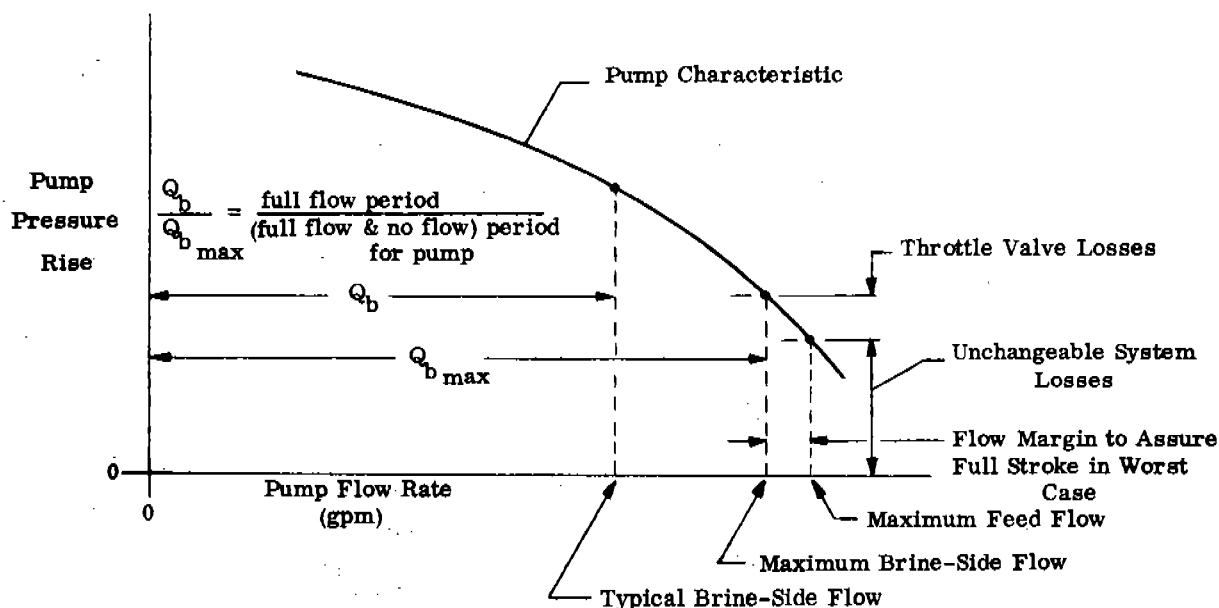


Figure 5-3: Pump Operation with the Simple Control System

## 5.4 Alternate Feed Pump Arrangements

The feed pump shown in Figure 5-2 is essential to the work exchanger system. The pressure rise developed by this pump is required in order to overcome the following losses:

- Line losses in the piping from the saline water source to the work exchanger
- Line losses in the brine discharge piping
- Flow losses and friction within the work exchanger itself
- Flow losses through the throttle valve in the control system

As shown in Figure 5-4, there are two ways to incorporate the feed pump within the reverse osmosis plant. In the arrangement shown in Figure 5-4A the feed pump is sized to handle a flow rate equal to the maximum waste brine discharge flow expected from the reverse osmosis system. In the arrangement of Figure 5-4B the feed pump must be sized for a larger capacity, a capacity equal to the desired saline water input flow rate to the membrane system.

The arrangement of Figure 5-4A is the one which has been used for the experimental work exchangers described in this report. The discussion of the operating characteristics shown in Figure 5-3 applies directly to this arrangement. The arrangement of Figure 5-4B employs a larger pump which might be part of a pretreatment system. The Figure 5-4B arrangement delivers flow to the main pump at an elevated pressure. This pressure may be desirable in order to meet the main pump NPSH requirements and assure that cavitation will not occur. The operation of the pump in the Figure 5-4B arrangement will differ somewhat from the previous description; the pump will have a continuous throughflow equal to the fresh water output flow rate from the membranes. Superimposed upon this steady flow delivery will be the on-off flow component associated with the operation of the control system.

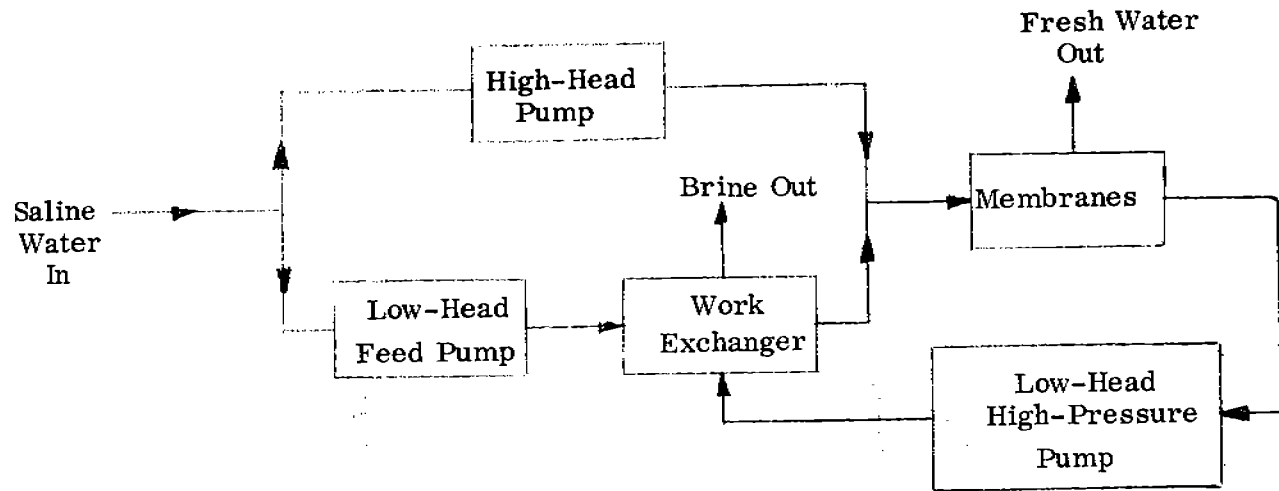


Figure 5-4A. Feed Pump for Work Exchanger Only

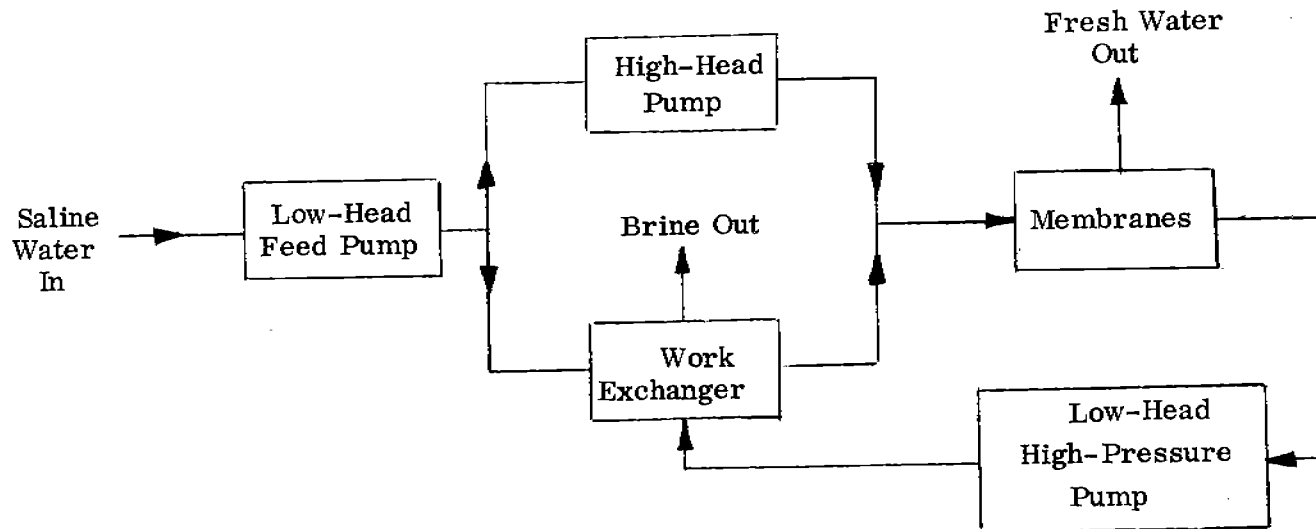


Figure 5-4B. Feed Pump Serves Complete System

## 5.5 Control System Design

One of the objectives of the second phase of this work exchanger test program was to construct and test an automatic control system for a selected type of work exchanger. A schematic diagram of the control system which was developed is shown in Figure 5-5. Flow switches are installed both in the high-pressure saline water pipe leading to the membranes and in the low pressure saline water feed pipe from the pump. These switches can be used in both bladder and piston type work exchangers to sense the end of flow delivery at the conclusion of each cycle.

The essential task for this control system is to determine whether the time interval between the end of the low pressure cylinder stroke and the end of the high pressure cylinder stroke is within the nominal range, too long, or too short. If the time interval is too short, an alarm is triggered to alert an operator to check why the manual throttle valve setting is no longer correct. (If the valve was set properly in the beginning, it should not have to be adjusted unless higher-than-expected brine-side flow rates are encountered or else the pressure losses in the low pressure side of the system increase due to fouling.) If the time interval is longer than a preset limit, an indicator light is turned on so that the operator is made aware of the situation and can readjust the throttle valve if the brine-side flow rate is likely to remain constant at that particular level for a long period. This adjustment is optional; its effect would be to minimize the time interval during which the feed pump flow is stalled.

A convenient way to sense time interval is to use a time delay relay which is triggered at a preset time interval after the low pressure bladder or piston reaches the end of its stroke. The relay will be reset without being triggered if the end of the stroke of the high pressure bladder or piston comes before the delay interval ends. If the stroke is not completed by the end of the delay interval, the alarm relay is actuated to indicate that the time interval is too long.

The flow switches and relays 1, 2, and 3 are actuated twice in each work exchanger cycle. If each cycle requires 12 seconds (6 seconds per stroke), the switches and relays will complete about 5 million operations per year in continuous service.

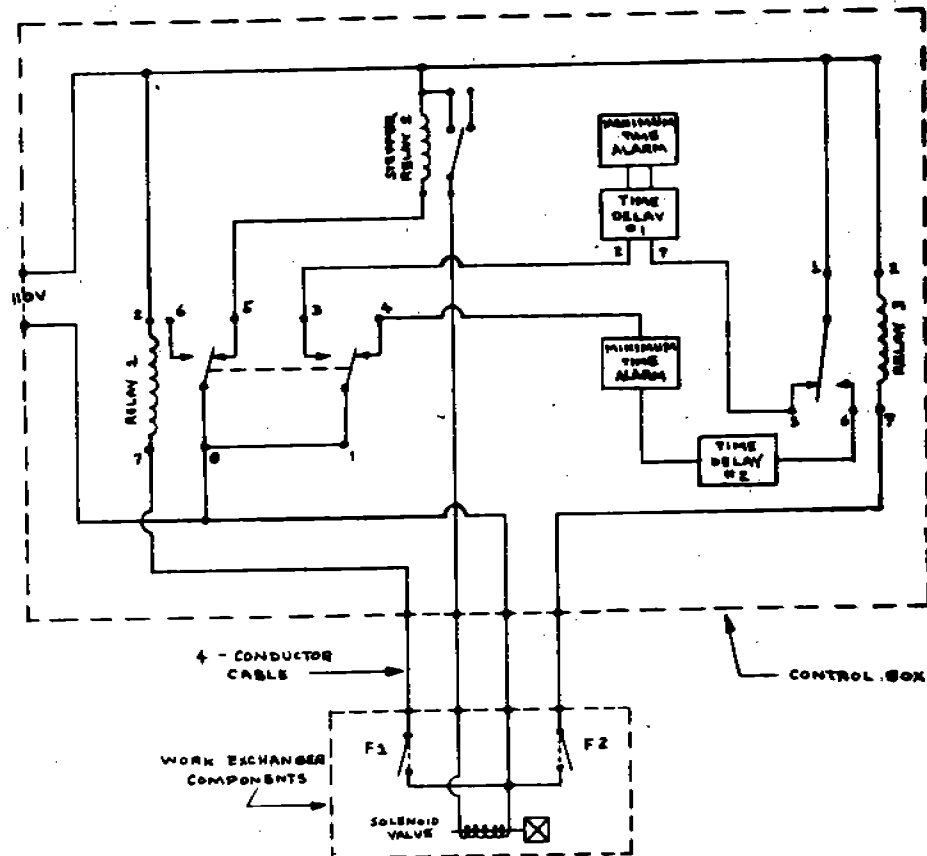
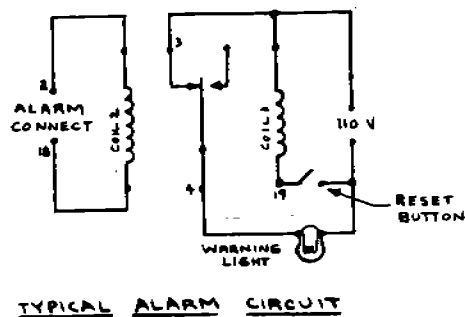
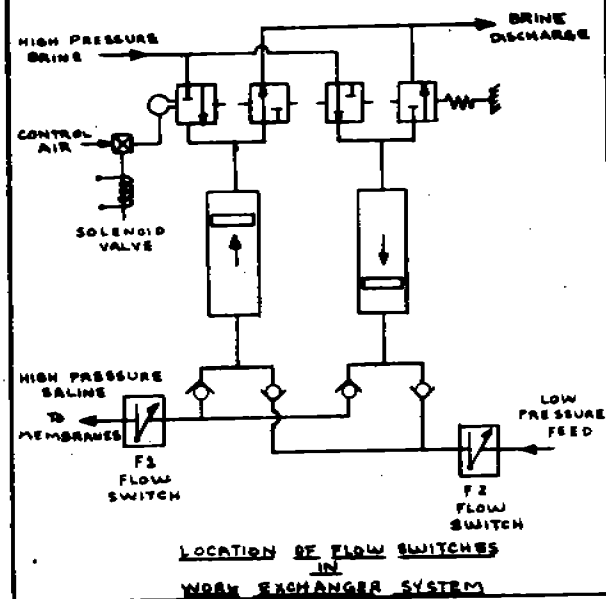


FIGURE 5-5  
WORK EXCHANGER CONTROL SYSTEM

Relays generally are rated by their manufacturers for 5 million cycles or more. They are easily replaced in the control system when necessary because they fit into standard plug-in sockets. The expected operating life of flow switches proved difficult for us to determine from the manufacturers. The principal electrical component is usually a microswitch, and its life expectancy generally exceeds 5 million cycles. The reliability of the mechanical components is not as well known.

Pressure switches which sense either the differential pressure change due to the presence or absence of flow in the saline water lines, or else the pressure difference across the individual work exchanger components, could be used as substitutes for the flow switches. Both types of pressure differences are largest when flow is being delivered by the feed line or cylinder, and fall off when the piston reaches the end of its stroke and flow stops.

Table 5-1 lists the particular components which were used in the automatic control system which we tested. The control system functioned properly except for two problem areas which were easily corrected.

- When the work exchanger was operated at less than one-half of the usual flow rate, the flow was inadequate to close the high-pressure flow switch F1. In this case, the control system would not cycle the work exchanger. This problem does not exist at the usual 10 gpm flow rate in the test work exchanger.
- After cycling occurs, the flow resumes more quickly in the low pressure side of the system than in the high pressure side. As a result, the minimum time alarm is actuated erroneously because the low pressure flow switch closes before the high pressure flow switch. The addition of a very short time delay in the low pressure flow switch circuit (i. e. , time delay #2) would solve this problem. This addition was not made in the prototype control system because it was not essential to the success of the test program.

Table 5-1

## CONTROL SYSTEM COMPONENTS

Component Designation in Figure 5-5	Manufacturer and Description
Relay 1	Sigma Instruments, Inc. 50 RO2-115 AC DPDT
Relay 2	Potter and Brumfield PC 11A 120 VAC DPDT
Relay 3	Sigma Instruments, Inc. 50 RO1-115 AC SPDT
Time Delay #1	Magnecraft Electric Co. W 211 ACP50X-5 DPDT operate delay, 1-10 sec., 120 VAC
Alarm Circuit Relays	Magnecraft Electric Co. W88 ALCPX-12 latching relay, 4 PDT
Flow Switch F1	McDonnell & Miller, Inc. #FS7-S-1 1 1/4" flow switch
Flow Switch F2	McDonnell & Miller, Inc. #FS4-3T1-3/4"

COMPARISON OF BLADDER AND PISTON TYPE  
FLOW WORK EXCHANGERS

6.1 Objectives and Scope of Comparison

A brief cost effective analysis of the two candidate flow work exchangers has been completed. The objectives of this analysis were the following:

1. To carry out an analysis to compare the relative technical and economic advantages of the two candidate work exchangers.
2. To provide information to guide the selection of the flow work exchanger unit to be tested with an automatic control system under Phase 2 of this program.

The areas considered in the comparison of the bladder and piston type flow work exchangers are the following:

1. Scale-up potential
2. Equipment reliability
3. Work exchanger efficiency
4. Flow work exchanger equipment costs

Each of these areas will be discussed in the following subsections.

6.2 Factors Considered in the Comparison

6.2.1 Potential for System Scale-Up

The heart of the flow work exchange system is the pair of pressure vessels in which the pressure exchange process takes place. Standard hydropneumatic accumulators for water service can be adapted for use as pressure vessels. Both piston and bladder type accumulators are commercially available to sizes of 80 gallon capacity using phenolic or chrome plated pressure vessels.

The piston units are easily adapted to have equally large ports on both ends of the cylinder. Larger size piston units could be made from long precision piping. The bladder units are very difficult to adapt for use with large ports on both ends. The standard bladder end of a transfer barrier accumulator is much too small to pass flow rates of more than 50 gpm without creating pressure drops greater than 20 psi. To pass several hundred gallons per minute, the molded bladder stem or the whole bladder assembly must be replaced, the pressure vessel hole enlarged, and the vessel pressure checked and certified. These modifications would be expensive and difficult to accomplish successfully in the bladder type units.

Other components such as ball valves and check valves are available in sizes at least up to 4 inches and 5 inches respectively in stainless steel with pressure ratings of 1500 psi. Appendix B shows that units consisting of two standard piston accumulators can supply up to 760 gpm or  $10^6$  gpd of feed water to a reverse osmosis system. Larger flow rates can be supplied if desired by several smaller systems operated in parallel.

#### 6.2.2 Reliability of Accumulators and Valves

The piston unit is basically more reliable than the bladder unit. A great deal of development work has been conducted on piston-type machinery. Several piston sealing techniques and many materials have been tested for various applications. Rubber O-rings are used on standard accumulator pistons, but cup seals and various ring seals could also be used. A piston seal may wear with usage and allow some leakage past the piston, but it will never fail. The leakage will be small because the pressure differences are very small.

Bladder units, on the other hand, can fail in several ways. The bladder continually pushes against a poppet valve once in each cycle causing excessive wear unless the flow is controlled to prevent the contact from occurring. The automatic control system would have to be much more complicated if the poppet valve is not moved at the end of each cycle. One of the bladders received from Kansas State University at the beginning of the present test program had 3 puncture holes near

the top. The new bladder purchased for this program developed six holes in the same area. These holes were caused either by punctures from the inside or by sticking of the bladder to the phenolic coating of the shell as a result of the elevated water temperature in our closed-loop system. The latter problem is believed to be the cause of the punctures. In any case, the bladder is more susceptible to complete failure which renders the work exchanger inoperable.

Check valves are conservatively rated for 10 million cycles which is equivalent to over 3 years at the highest practical cycling rate, 5 cpm. Ball valves will probably require replacement of seals as often as every 500,000 cycles which is equivalent to about two months at the highest practical cycling rate. However, these seals can usually be replaced quickly and replacement can be scheduled at regular maintenance intervals.

### 6.2.3 Efficiency of Flow Work Exchangers

The analysis presented in Appendix B resulted in calculated levels of efficiency for piston type flow work exchangers over a range of flow capacities. The results of that analysis are shown on Figure 6-1. Efficiency levels are shown to range from as low as 65% to as high as 99%. High efficiencies are attained by selecting large components which yield very small pressure losses so that the associated power required by the pumps is small. Details of the equipment size are given on Table B-2. The cost of the equipment is presented on Figure 6-1 which shows that high efficiency requires more expensive equipment.

The units 1, 2, 3, and 4 on Figure 6-1 refer to Table B-2 and correspond to units based upon standard piston accumulators of the following sizes: 10, 20, 40, and 80 gallons. Units 3A, 3B, and 3C are all sized for a 40 gallon accumulator but employ different sizes for the other components.

The control valves used with the flow work exchangers for the large flow rate units must be ball valves instead of four-way, two-position valves. The ball valve units have very low pressure loss. The four-way valves have very large pressure losses in the same port sizes. For instance, unit 3Ba on Tables B-2

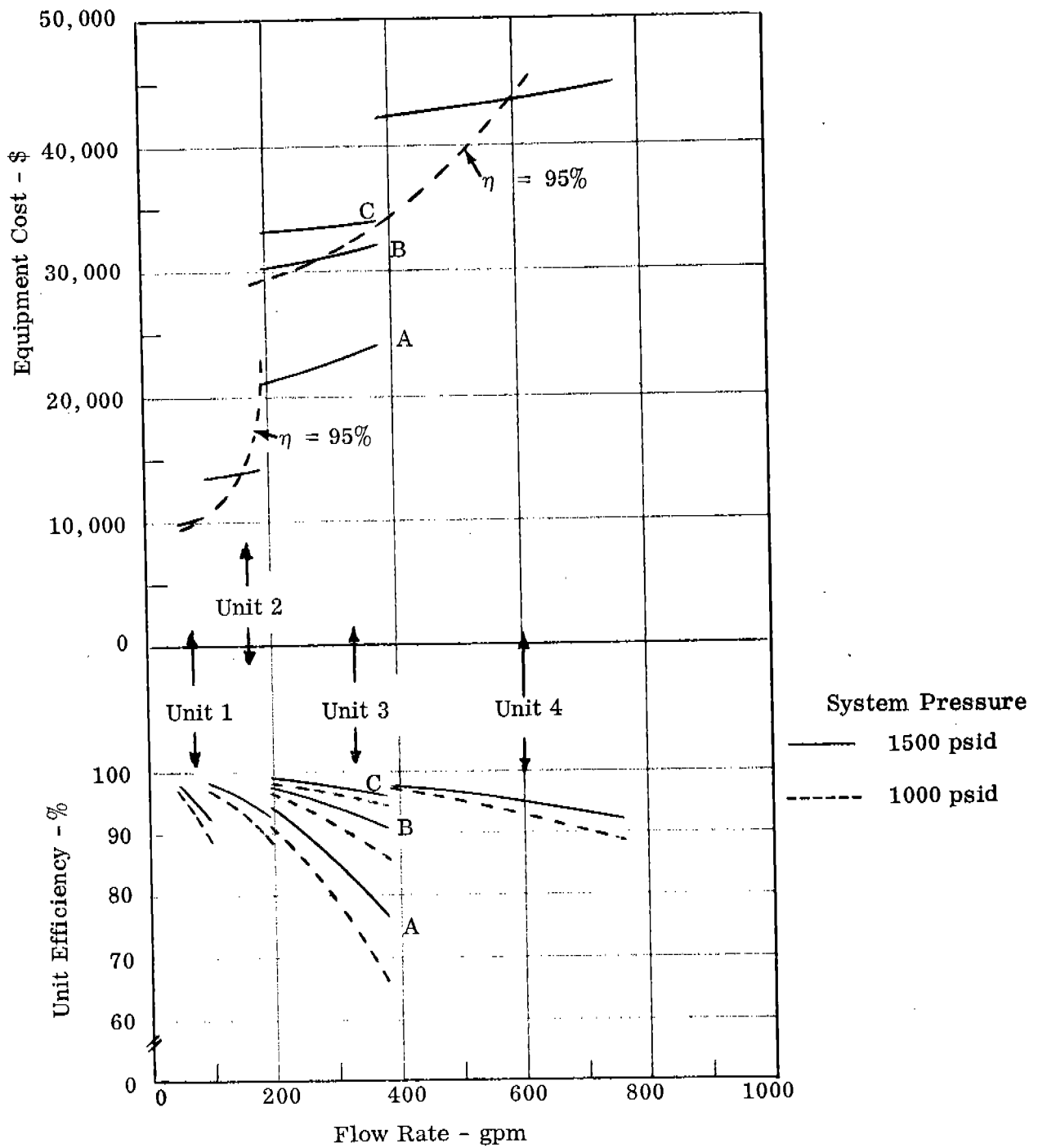


Figure 6-1. Piston Type Flow Work Exchanger Cost and Efficiency vs. Flow Rate

and B-3 loses only 3.5 psi across a 2-1/2" ball valve at 380 gpm. A four-way valve of the same port size would lose 150 psi and a 6" valve would lose 15 psi. Not only would the efficiency be significantly reduced for large sizes, but the cost would also be significantly larger. Four-way valves can be used on the small flow rate units without significant efficiency penalties and offer the advantage of very long cycle life (10 to 20 million cycles).

Two very important pressure losses are not included in the calculated flow work exchanger efficiency. These losses are:

1. The pressure loss in the high pressure brine and feed water lines attaching the flow work exchanger to the reverse osmosis system.
2. The brine-side pressure loss in the reverse osmosis membrane system.

Figure B-9 shows that each additional 50 psi of pressure loss in either the brine flow piping or the feed flow piping will result in about 8 points loss in efficiency for a 1000 psid flow work exchanger. The brine-side membrane pressure drop should not be included in the flow work exchange efficiency calculation, but the head rise and cost of the high pressure pump are affected by this loss.

The pressure losses and efficiencies of the bladder and piston units would be equivalent if the bladder port is properly modified. If the bladder port is too small, the pressure loss can be calculated from Eqs. (B.2) and (B.3) or approximated by taking half of the losses shown on Figure B-4. The efficiency loss can then be estimated from Figure B-9. For instance, unit (3Bc) with a one inch modified port would have a pressure loss of approximately 50 psi/2 or 25 psi. This would result in 4 points loss in efficiency for a 1000 psid system. At larger flow rates the penalties are much more severe unless the port size is increased adequately.

#### 6.2.4 Comparative System Costs

The flow work exchanger system equipment costs are shown as a function of size on Figure 6-1. The unit designations refer to the equipment sizes, materials, and performance listed on Tables B-2 and B-3. The costs shown on

Figure 6-1 apply only to the cost of the unmodified equipment. No allowances have been included for welding or installation costs.

A line of constant efficiency has been included on the cost curves. The line is not a continuous curve, primarily because of the change from threaded to flanged ball valves between unit 3A and 3B. In a range of flow rates between 200 and 300 gpm it may be more economical in terms of equipment cost to operate 2 small size units rather than one large unit. For units larger than 300 gpm, Figure 6-1 shows that one single unit will have a lower cost than several small units.

The bladder accumulators are more expensive than piston accumulators as shown on Figure B-6. In addition, the bladder units require major modifications to the bladder stem and shell to increase the port size. In first cost alone, the bladder type accumulator is 2 to 3 times the cost of the piston type accumulator. For unit 3Ba (Table B-2), a bladder type work exchanger would cost \$5,800 more or about 18% more than a piston type flow work exchanger. With the cost of the modification to the bladder accumulator included, the total cost increase would be 20% to 25% above that of a piston type unit.

### 6.3 Cost Effective Analysis - Summary and Conclusion

A summary of the comparison of the two types of flow work exchange units is shown on Table 6-1. Figure 6-1 shows the detailed efficiency and equipment cost comparison for the range of units investigated. On the basis of this analysis, the piston type flow work exchanger system is superior to the bladder unit for this application. Therefore, the piston unit was selected for test with the automatic control system under Phase 2 of this program. This decision was based upon the following considerations:

1. The piston type unit is available in standard sizes with ports that will accommodate large flow capacities without major modification of the unit. The bladder units require major modification in all sizes. Piston units larger than 80 gallons could be fabricated from precision pipe.

Table 6-1

## Comparison of Flow Work Exchange Units

Area of Comparison	Bladder-Type Flow Work Exchanger	Piston-Type Flow Work Exchanger
Potential for System Scale-up to Larger Capacities	<ol style="list-style-type: none"> <li>1) Standard accumulators are available up to 80 gallons in size. On end has a port size of 2 to 4 inches. The other end always has a port size less than 3/4" in diameter.</li> <li>2) The small port causes large pressure drops at flows over 50 gpm. Enlargement of this opening is expensive and very difficult to accomplish successfully.</li> </ol>	<ol style="list-style-type: none"> <li>1) Standard accumulators are available up to 80 gallons in size with 3 inch ports on both ends. These ports can also be enlarged easily if required.</li> <li>2) Larger sizes could be fabricated from precision pipe or tube of any desired length.</li> </ol>
Reliability of Standard Accumulators for Use in Flow Work Exchange Units	<ol style="list-style-type: none"> <li>1) Bladders will fail completely due to wear at the poppet valve contact point or possibly on the vessel sides.</li> <li>2) Bladders are also susceptible to punctures as experienced in our test program.</li> </ol>	<ol style="list-style-type: none"> <li>1) Piston units are basically more reliable because the seal will only wear but never fail.</li> </ol>
Efficiency of Flow Work Exchange Units	<ol style="list-style-type: none"> <li>1) The standard bladder-end port size is very small and cannot easily be enlarged enough; the efficiency will be reduced significantly in comparison to that of a piston unit, especially at high flow rates.</li> </ol>	<ol style="list-style-type: none"> <li>1) The piston accumulators and their ports are easily supplied in large sizes for low pressure losses at high flow rates.</li> <li>2) Flow work exchanger efficiencies above 90% are easily attainable at all flow rates investigated.</li> </ol>
Comparison of System Cost as a Function of Capacity, Efficiency, and System Pressure	<ol style="list-style-type: none"> <li>1) Bladder-type accumulators cost 2 to 3 times more than piston-type accumulators.</li> <li>2) Bladder-type units require major modifications to the bladder assembly and shell.</li> <li>3) The first cost plus modification cost make a bladder-type flow work exchanger 20% to 25% more expensive.</li> </ol>	<ol style="list-style-type: none"> <li>1) Flow work exchanger unit costs for piston-type units are shown on Figure 6-1. Piston-type units are significantly less expensive than bladder-type units.</li> </ol>

2. The piston accumulator is more reliable than the bladder unit. The bladder is susceptible to failure which requires shutdown of the work exchanger.
3. The two types of units would have equal efficiencies if the bladder port is enlarged to a size equal to that of the piston unit. Otherwise, bladder unit efficiencies would be significantly lower than piston unit efficiencies.
4. Development work under Phase 1 determined that the piston unit can be modified more easily than the bladder unit to bleed air from the feed water side of the system (see Section 4).
5. A bladder type flow work exchanger costs approximately 20% to 25% more than a piston type work exchanger.

## Section 7

### RELIABILITY TESTING PROGRAM FOR A PISTON TYPE WORK EXCHANGER

The objectives of the reliability testing program were the following:

- To demonstrate the performance of the control system,
- To establish that the work exchanger is operable with the selected control system,
- To obtain performance reliability information for the control system and work exchanger components.

The testing program was divided into a series of four test periods of one month duration and five equipment inspections. During each inspection and test period, the following components were evaluated:

#### Inspections

1. The piston seals were inspected.
2. The accumulator walls were inspected.
3. The piston seals and check valves were leak checked.
4. A check valve was inspected at the completion of testing.

#### Test Period

1. The operation of the check valves and the control valve are monitored.
2. Operation of the automatic control system was evaluated.
3. Records of component failures were maintained.

The piston type work exchanger was selected for operation with the automatic control system in the reliability test program. In Section 6, the piston type unit was shown to be better than a bladder type unit on the basis of scale-up potential, reliability,

and cost. The piston work exchanger was operated on fresh water at 800 psig system pressure and a flow rate of about 8 gpm for the whole test program. The unit cycle time was about 36 seconds (up and down).

## 7.1 Experimental Observations

The automatic control specified in Section 5 operated successfully throughout the reliability test program. The work exchanger is easily started-up and set into automatic cycling with full piston strokes in less than a minute if the system is full of liquid. Startup requires about 5 minutes if time is needed for bleeding air from the system. The unit will continue automatic cycling without adjustment of the low pressure flow rate for long periods of time.

During the four test periods, three different piston sealing designs were tested and evaluated. The complete work exchanger unit was operated for a total of 619 hours which represents approximately 60,000 component cycles (up and down for pistons, open and close for valves). A summary of test observations is presented on Table 7-1. A discussion of specific components and problem areas is presented in the next section.

## 7.2 Discussion of Specific Components

### 1. Failure of the First Cup Piston

86 hours after the start of the first test period, the work exchanger stopped cycling due to leakage of high pressure flow past the cup piston (see Figure 4-3). The cup piston was removed from the cylinder and inspection revealed that the seals failed due to the squeezing of the seals by pressure forces exerted on the piston at the end of each stroke. To prevent these large end forces on the cup seals, four metal rod extensions were installed on each end of the piston. This solved the seal failure problem, so that a new cup seal piston could be operated successfully for 502 hours during the remainder of the test program.

Table 7-1  
RESULTS OF WORK EXCHANGER RELIABILITY TESTING

Equipment Inspection	Number of Test Hours and Cycles	Pistons Tested and Cumulative Hours and Cycles for Each (1 cycle is up and down)	Condition of Pistons and Seals	Condition of Accumulator Walls and End Caps	Other Observations During Inspection or Test Period
Initial Inspection	Per Test Period	O-ring Piston (see Figure 4-3)	New O-rings were installed on a standard anodized aluminum piston.	<u>Left Accumulator</u> Very good condition. No scratches. Only 1 pit, about 1/16" diameter, was visible through the phenolic coated wall.	1. There was no significant leakage past the check valves or the piston seals.
	Cumulative	Cup Piston (see Figure 4-3)	Two cup seals used in the Phase 1 testing were installed on one piston. The side of the cups were creased circumferentially as though they had been bent but showed no visible wear.	<u>Right Accumulator (No. 1)</u> One section of phenolic coating was missing from the wall and the surface was badly pitted. (3" ring, 5" in one 3" section of circumference.) Several other smaller spots could also be seen.	
Following First Test Period	117 hours 11,300 cycles	O-ring piston 117 hours 11,300 cycles	The O-rings showed visible wear including flat spots and peaked points on the O-ring. The piston side surfaces did not touch the walls during this test period.	<u>Left Accumulator</u> Condition is worse. There are now about 20 full length axial scratches and a large number of shorter ones, but only a few are through phenolic. A ring of 30 pits has also appeared, ranging in size from .015 to .060 inches.	1. There were no difficulties experienced in bleeding the system of air during startup. 2. No check valve, 4-way control valve or pump malfunctions occurred. 3. The automatic control operated properly.
	117 hours 11,300 cycles	Cup Piston 86 hours 8,360 cycles	The cup seals failed after 86 hours of testing because of large pressure forces which squeezed the seals at the end of each stroke. A second O-ring piston was installed for the remaining 31 hours. The O-rings and piston surfaces were worn away by the rough walls.	<u>Right Accumulator (No. 1)</u> The walls are significantly worse. The area noted before has enlarged slightly and a 3" wide axial section has about 75% of the phenolic coating removed. This accumulator can no longer be used.	
Following Second Test Period	153 hours 14,800 cycles	O-ring piston 270 hours 26,100 cycles	Teflon back-up rings were placed in each O-ring groove. At the end of the test period, the O-rings showed no increased wear.	<u>Left Accumulator</u> Condition is worse. Two of the axial scratches recorded above are now pitted along most of cylinder length. The ring of pits has increased to about 42 major pits plus more minor pits.	1. There was no significant leakage past the check valves or the piston seals at the beginning or at the end of the second test period. 2. No check valve, 4-way control valve or pump malfunctions occurred. 3. The automatic control operated properly.
	270 hours 26,100 cycles	Cup Piston 153 hours 14,300 cycles	A new cup piston was installed with four extended rods on each end of the piston to protect the seal against the large end loads. These rods were bent by the end loads too. The cup seals showed no noticeable wear during the test period.	<u>Right Accumulator (No. 2)</u> A spare accumulator was installed to replace the damaged one. It had no pits or scratches at the beginning of the test period or at the end of the test period.	
Following Third Test Period	145 hours 14,100 cycles	Polypak Seal Piston (see Figure 7-1) 145 hours 14,100 cycles	The piston was very tight in the cylinder. The seals on both ends showed 23 axial grooves worn in the material by 23 corresponding scratches in accumulator. The grooves were about 1/16" wide and .005 to .010 inches deep.	<u>Left Accumulator</u> Condition is worse. About 30 axial scratches are visible and 9 of them are pitted into a metal. The pitted ring mentioned before is almost a solid ring now.	1. There was no significant leakage past the check valves or the piston seals at the end of the test period. 2. No check valve, 4-way control valve or pump malfunctions occurred. 3. The automatic control operated properly. 4. The pressure drop for the polypak seal system was 25 psi as compared to 7 psi for the cup seal at 7 gpm.
	415 hours 40,200 cycles	Cup Piston 298 hours 28,900 cycles	The extended rods were reinforced to eliminate the bending. No axial scrapes appeared on the seal or piston and the cup seal showed no added creasing. Two 1/16" shiny rings did appear on both cups and the sharp lip was beginning to fray.	<u>Right Accumulator (No. 2)</u> Four pits about 1/16" in diameter are now visible on wall. No axial scratches are visible.	
Final Inspection Following Fourth Test Period	204 hours 19,800 cycles	Polypak Seal Piston 349 hours 33,900 cycles	The seals on both ends showed about 58 axial grooves worn in the material by the accumulator scratches. The new orientation of the piston to the scratched wall produced the new scratches. The whole 0.55" axial length of the seal was shiny and worn.	<u>Left Accumulator</u> Condition is worse. There are still about 30 axial scratches but now 19 of them are pitted into the metal. The ring of pits is worse but still not quite a solid ring. (Width of pitting is about 1/16")	1. The check valves and piston seals allowed no leakage. 2. No check valve, 4-way control valve or pump malfunctions occurred. 3. The polypak seal piston pressure drop was reduced to about 21 psi by removing 0.04" from the seal groove inner diameter. At the end of the test period the pressure drop was reduced to about 16 psi at 7 gpm due to seal water. 4. One relay in the automatic control was not operating properly, causing the work exchanger to stop cycling occasionally. 5. The air solenoid valve was beginning to hum loudly at the end of the test period.
	619 hours 60,000 cycles	Cup Piston 502 hours 48,700 cycles	The cup seals show no significant change in condition.	<u>Right Accumulator (No. 2)</u> Twelve pits are now visible, the largest are about 0.10". No axial scratches are visible.	

## 2. Seal Wear Due to Rough Cylinder Walls

The right side accumulator used during the first test period had a rough pitted ring section of wall about 3" to 5" long where the phenolic coating had been removed. At the end of the test period, there was also a 3" wide section of pitted wall extending almost the full length of the accumulator where 75% of the phenolic was removed. The rubber O-ring seals on the standard piston (see Figure 4-3) used in this accumulator for the last 31 hours of the first test period were worn flat by the 3" wide axial pitted area to the point where the piston scraped on the wall. The remaining 270 degrees of the piston O-rings also showed moderate wear but the piston did not scrape the wall.

The axial pitted grooves that developed in the left accumulator scratched corresponding grooves in the polypak seals.

For successful long-life operation of piston seals, it is important that the cylinder wall be kept in a smooth condition. This will require a good cylinder wall plating or a solid wall material compatible with salt water and careful filtering of the incoming feed and brine flows.

## 3. Deterioration of Accumulator Walls

The accumulators used in the fresh water test unit were standard water service accumulators with phenolic coated walls. The phenolic coating will not be suitable for salt water application because of the ease with which it is scratched and its susceptibility to penetration by tap water.

The left accumulator used throughout the test program developed a large number of axial scratches from small particles trapped and pushed by the piston seals. Once formed, these scratches deteriorated into pitted grooves over the course of the testing program.

Two of the accumulators developed ring shaped pitted areas around the full circumference of the accumulator. The first right-hand accumulator already had a substantial ring shaped pitted area when received from Kansas State University, apparently caused by leaving the piston in one position for several weeks with some water still in the accumulator. The left accumulator developed a thin ring of pits (about .060" wide) during our test program which apparently started at one of the piston seals when it was left stationary for a period of 3 or 4 days.

To protect the accumulator walls in future units, the liquid should be filtered with a 10 micron nominal filter and the unit should be drained and dried when it is to be left idle for more than a few days.

4. Comparison of Measured System Pressure Losses for Three Pistons

The pressure loss for the low pressure flow ( $\Delta P_l$ ) and the high pressure flow ( $\Delta P_h$ ) is measured by the differential pressure gages shown on Figure 4-2. The following listing is a comparison of these measured losses using the three different types of pistons at a system flow rate of 7 gpm. The difference between the listed pressure losses is due to the increased frictional resistance of the different pistons.

<u>Piston</u>	<u>System Pressure Loss at 7 gpm</u>
Cup Piston	7 psi
O-Ring Piston	10 psi
Polypak Piston Original	25 psi
Polypak Piston with .040" removed from seal groove I. D. (beginning of test period)	21 psi
Polypak Piston with .040" removed from seal groove I. D. (end of test period)	16 psi (reduced due to seal wear)

5. O-ring Seal Piston

The Buna-N O-ring seals performed satisfactorily for 270 hours in a smooth walled accumulator. However, the standard rubber O-ring seal configuration does not appear to be satisfactory for long service operation under work exchanger operating conditions.

The rate of wear and rolling of the seal was reduced by using a teflon back-up ring during the second test period. When the walls become pitted and rough, the Buna-N O-ring wears very rapidly. EPDM rubbers may give longer service life. Rubber seal cross-sections other than circular may give better service for work exchanger operating conditions.

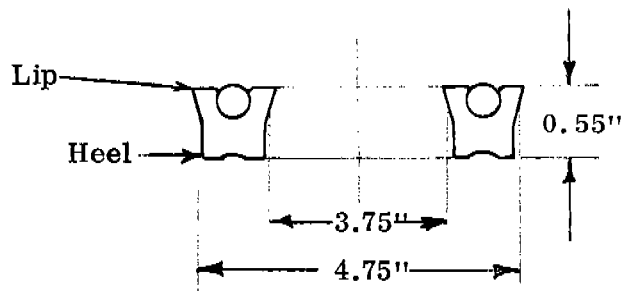
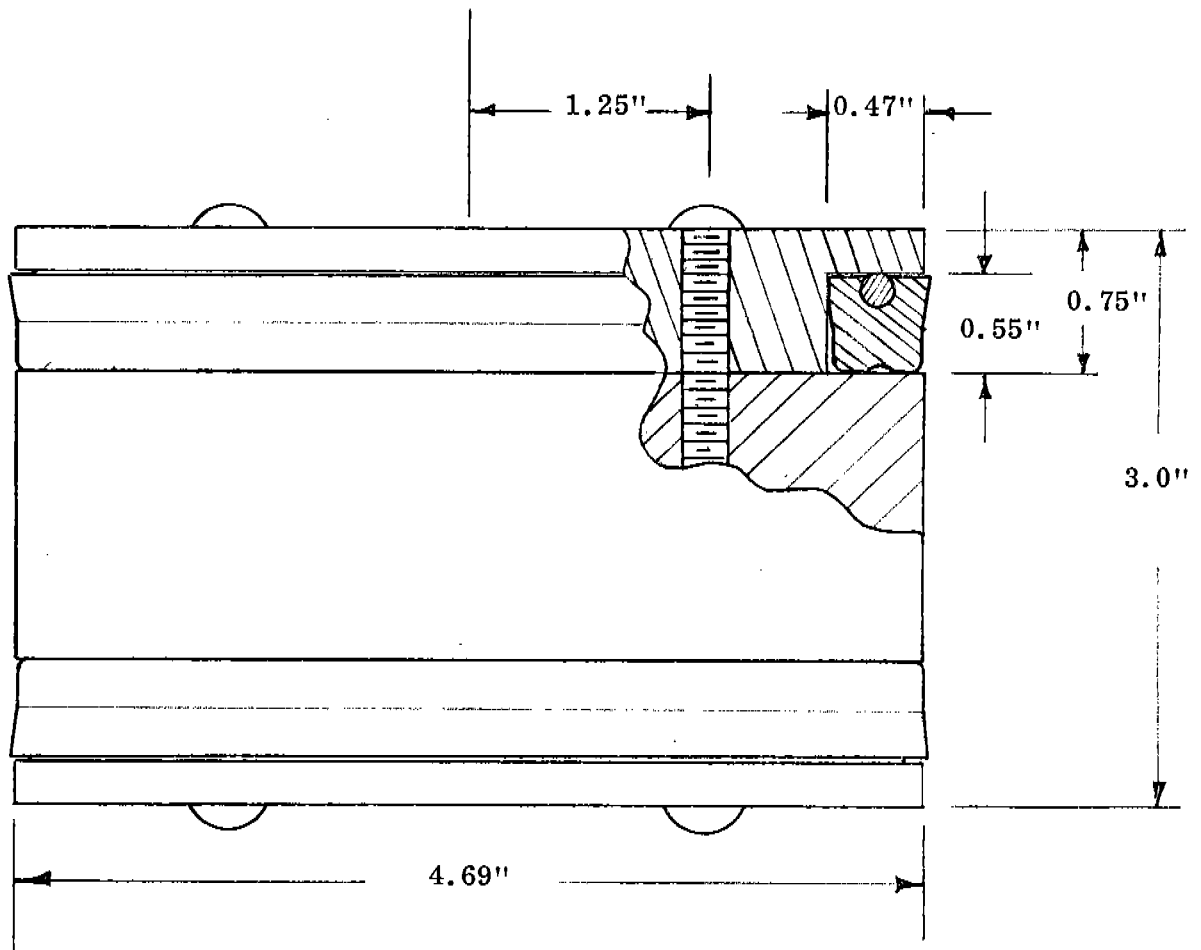
6. Cup Seal Piston

The cup seal made of molythane material (see Figure 4-3) was tested for 502 hours with very little wear to the seals. The system pressure drop when using this piston was also the smallest as discussed in 4 above. Of the three pistons tested, this piston will give the best service under work exchanger operating conditions.

7. Polypak Seal Piston

The polypak seal made of molythane material was tested for 298 hours with unsatisfactory performance. The seal cross section and piston configuration are shown on Figure 7-1.

The proper polypak seal and correct seal groove dimensions resulted in the piston being very tight in the cylinder. This caused the pressure difference required to push the piston to be about 18 psi larger than that required for the cup seal piston.



Polypak Seal  
Molythane Material  
Parker Seal Co.

Figure 7-1. Polypak Piston and Seal

Wear of the seal and an enlarged seal groove (.040" deeper) reduced this value to 9 psi larger than the cup seal. Axial scratches on the seal made by pitted scratches on the cylinder walls cut into both the lip and the heel of the seal indicating that practically the whole side surface of the seal touches the wall. In fact, at the end of 298 hours, almost the whole side surface of the seal was shiney from wear except for the axial scratches.

From the experience of this test program, the polypak seal configuration would appear not to be satisfactory for the operating conditions of the work exchanger.

#### 8. Automatic Control Operation

The control system functioned as designed to automatically alternate the two cylinders to accept and supply a continuous flow of high pressure water. The design and operation of the control system are presented in Section 5.

The control system operated satisfactorily for a total of 547 hours which is equivalent to about 53,000 cycles. During the last 40 hours of testing, trouble developed in relay number 2 (see Figure 5-5 and Table 5-1). Once or twice a day, the work exchanger would stop cycling. The cause of this problem was traced to relay number 2 which would occasionally stick in one position or the other. The system could be started again by turning the control system switch off and on once. When this relay was replaced, the control operated properly again. More reliable relays to replace relay number 2 are now available from the same manufacturer. Future control systems should use the new enclosed relay.

The switching time transient observed on the high pressure gauges and flow meter was increased from about 0.3 seconds

to about 0.6 seconds when changing from the mechanical timer control to the automatic control. The reason for this is that the high pressure flow rate must almost stop before the flow switch will respond to actuate the automatic control system whereas the timer system would switch cylinders before the piston reached the end of the cylinder. The increased switching time is not considered to be significant, since it is still a small percentage of the cycle time.

#### 9. Check Valve Wear

No operational problems occurred during the test program. At the end of 619 hours of testing (60,000 cycles), one of the check valves was disassembled for inspection. The valve used was a Circle Seal, 3000 psi, 3/4" brass piston type check valve. The valve seals, O-ring seal, and spring were all in excellent condition. The sliding piston showed a small amount of wear, but it represented only a very small percentage of the piston life (1% to 5% estimated). Therefore, the piston type check valve may be expected to give very good performance under the work exchanger operating conditions.

### 7.3 Conclusions

The results of the reliability test program led to the following conclusions concerning the automatic control and the work exchanger components:

1. The automatic control system designed for the work exchanger does properly and reliably cycle the two work exchanger cylinders to accept and supply a continuous flow of high pressure water.
2. A phenolic coating on accumulator walls will not be satisfactory for work exchanger operating conditions.

3. To prevent pitting and scratching of accumulator walls, the low and high pressure flows should be filtered ( $10\mu$  nominal) and the system should be drained and dried when it is to be idle for more than four days.
4. Badly pitted accumulator walls cause seals to wear rapidly and unevenly, significantly reducing the seal cycle life.
5. The cup type piston seal was the best of three piston seals tested in this program (others were O-ring and polypak). The cup seal was tested for 502 hours (48,700 cycles) with very little seal wear and the smallest piston frictional resistance.
6. The check valves, 4-way valve, the low pressure pump, and the submersible pump operated satisfactorily for the complete 619 hour test program without any malfunctions.

Section 8  
CONCLUSIONS

The following conclusions were reached as a result of the work completed under this program:

- (1) Efficiencies in the range of 90% to 97% were calculated for the prototype work exchangers based upon measured pressure loss data for system pressures above 1000 psig. Losses in the external piping required to connect the work exchange unit to a reverse osmosis plant will reduce the actual efficiency below these values. The magnitude of the efficiency loss can be limited by using large pipe sizes.
- (2) Flow work exchange units can be built in sizes up to 600,000 fresh-water gpd at 35% recovery in reverse osmosis systems. Efficiencies of these units will be within the range of 90% to 95%. Units in parallel can be used to achieve capacities larger than this.
- (3) A piston-type flow work exchanger unit is better than a bladder-type system because the piston unit is more easily modified for use, is more reliable, is more easily bled of trapped air, and is 20% to 25% lower in cost.
- (4) An automatic control system using flow switches or pressure switches to sense when the piston or bladder reaches the end of its stroke can be used in conjunction with a manual control valve and alarm system to control either the piston-type or bladder-type flow work exchanger. The low pressure feed flow is adjusted to be slightly larger than the high-pressure flow and the sensing system alerts the operator if the system operation gets out of adjustment. Use of a more sophisticated control system would only be warranted after a thorough analysis of the complete reverse osmosis plant operation.

- (5) An automatic control system was designed, installed and successfully operated with a piston type work exchanger for 547 hours (53,000 cycles).
- (6) The accumulator walls and the piston seals showed a great deal of wear during the test program. As a result, phenolic coated accumulator walls are not considered suitable for work exchanger service. A cup type piston seal was the best of three types of piston seals tested.
- (7) Inlet water flows to the work exchanger should be filtered (10 $\mu$  nominal) to protect cylinder walls and piston seals.
- (8) The check valves, the 4-way valve and two system pumps operated without malfunction for 619 hours (60,000 cycles).

## RECOMMENDATIONS

The flow work exchanger is a very efficient device for energy recovery from the brine discharge of a reverse osmosis plant. Further investigations are needed to improve critical component life, reduce critical component replacement cost and maintenance, and determine the effect of the use of flow work exchangers on fresh water cost. Recommendations of specific areas for further investigation are the following:

- (1) Life testing of ball valve and other low pressure-loss valves for handling flow rates of 200 to 800 gpm should be undertaken. The high efficiency of a flow work exchanger operating in this flow rate range requires the use of low-pressure-loss control valves. The cycle life of these valves under work exchanger operating conditions has been estimated to be only 1 to 3 months by several manufacturers.
- (2) The two pressure transfer piston cylinders are the heart of the work exchanger system. To achieve component life times compatible with a reverse osmosis system (30 years), the cylinders will have to be made from an expensive material unaffected by saline water or the cylinders will need inexpensive replaceable liners. An analysis, design, and testing program should be undertaken to determine how to maximize the cylinder life and minimize the equipment cost over the life of a reverse osmosis plant (30 years).
- (3) An economic analysis of a reverse osmosis plant including a flow work exchanger energy recovery device should be undertaken. The flow work exchanger has two outstanding economic advantages in its favor:

- The energy recovery efficiency of a work exchanger in a reverse osmosis plant is almost double the turbine-generator-motor-pump combination normally used.
- The use of a work exchanger will reduce the required reverse osmosis high pressure feed pump capacity by a factor of 1.5 to 2. The cost of this pump will, therefore, be correspondingly lower.

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## DERIVATION OF PERFORMANCE EQUATIONS FOR FLOW WORK EXCHANGERS

### A.1 Ideal Energy Exchange Between Two Liquid Streams

The purpose of the flow work exchanger is to transfer energy and pressure from a high pressure waste stream to a low pressure feed stream. Assuming that there are no losses associated with this process, the pressure and volume changes in the exchanger pressure vessels can be shown on an ideal pressure-volume indicator diagram. The following assumptions are made in order to draw and evaluate the ideal indicator diagram:

#### Assumptions

1. All process steps are isentropic. There are no fluid friction losses in the pipes and valves, and no emptying and filling losses in the pressure vessels.
2. There are no frictional losses due to the sliding pistons.
3. The compressibility of the water is neglected in the analysis. In reality, water will compress about 0.77% at 1500 psi.
4. Leakage flow through valves and past pistons is assumed to be negligible.

The ideal energy exchanged between the high pressure brine and low pressure feed during one complete cycle of each pressure vessel is represented by the area 1-2-3-4 on Figure A-1. The value for the ideal power transferred for two cycling vessels is given by the following equation:

$$W_i = (P_h - P_\ell) (V_1 - V_2) \frac{2 N(144)}{60}$$

or

$$W_i = (P_h - P_\ell) (Q_h)(0.321) \quad (A-1)$$

Figure A-1  
Ideal Indicator Diagram  
for a Flow Work Exchanger

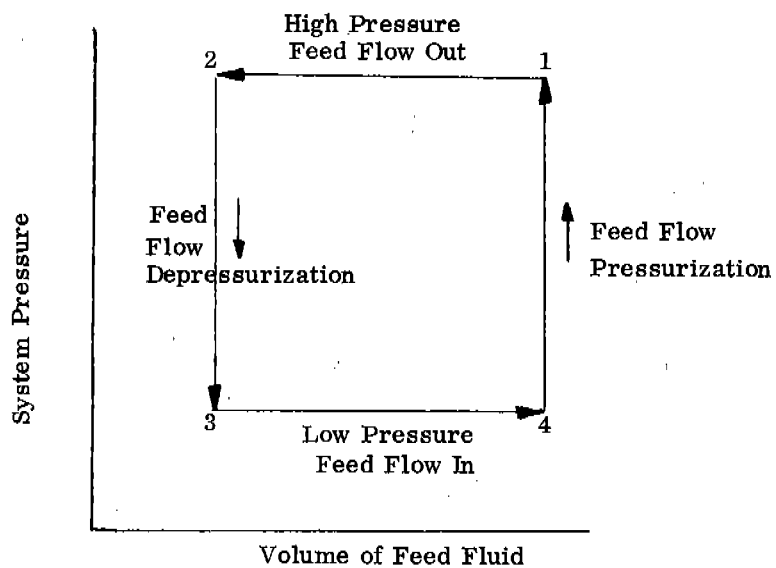
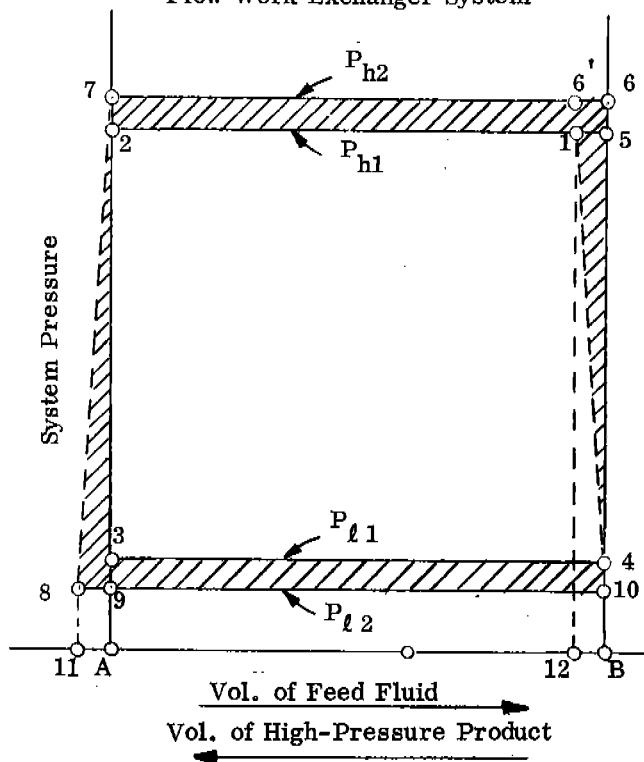


Figure A-2  
Indicator Diagram for Actual  
Flow Work Exchanger System



## A.2 Losses in an Actual Flow Work Exchange System

Losses occur in an actual system which make the efficiency of the energy exchange less than 100%. These losses can be evaluated from measured fluid flow losses or from the power additions required to operate the flow work exchanger. Table A-1 lists and describes the system losses in both ways. Figure B-1 shows the components required in the operation of a flow work exchanger in a reverse osmosis system. Power must be supplied to two pumps, automatic controls, solenoid valves and pneumatic systems. Each of these power inputs represents a loss in the flow work exchange system. The efficiency of the flow work exchanger is defined as the net power transferred divided by the ideal power transfer from the brine flow to the feed flow.

$$\eta = 1 - \frac{W_{\ell} + W_h + W_c + W_a}{W_i} \quad (A-2)$$

The power input to the low pressure pump is calculated by the following equation:

$$W_{\ell} = (1 - F_{\ell}) W_{ocl} + F_{\ell} (W_{tl}) \quad (A-3)$$

In this equation the first term represents the power input during the normal operating condition and the second term represents the power input during the transient switching condition. The low pressure pump input power can also be calculated from the following equation using measured pressure losses and flow rates.

$$W_{\ell} = (1 - F_{\ell}) \frac{\Delta P_{\ell} Q_{\ell}}{\eta_p \eta_m} (.321) + F_{\ell} W_{tl} \quad (A-4)$$

A similar equation can be written for the work input to the high pressure, low head pump.

$$W_h = (1 - F_h) W_{och} + F_h W_{th} \quad (A-5)$$

Table A-1

## FLOW WORK EXCHANGER SYSTEM LOSSES

Power Input Symbol	Power Input Description	Fluid System Losses Requiring Supply of Power	Test System Measurements	Losses to be Estimated for Incorporation of Flow Work Exchanger into Reverse Osmosis System
$W_l$	Power input to low pressure seawater feed pump	<ol style="list-style-type: none"> <li>1. Friction and expansion losses in pipes, fittings, check valves, and control valves during feed and exhaust part of cycle.</li> <li>2. Filling and emptying losses at inlet and outlet ports of accumulators.</li> <li>3. Power input when control valve is switching and flow is momentarily stopped.</li> <li>4. Frictional resistance to piston movement.</li> <li>5. Transient expansion and flow losses during depressurization.</li> </ol>	<ol style="list-style-type: none"> <li>1. <math>\Delta P_l</math> - measured pressure loss from check valve inlet to control valve exit.</li> <li>2. <math>P_l</math> - feed pressure.</li> <li>3. <math>Q_l</math> - low-pressure flow rate.</li> <li>4. Fraction of cycle corresponding to no-flow operation.</li> </ol>	<ol style="list-style-type: none"> <li>1. Friction pressure loss of exhaust line piping (this is power which would not have to be supplied in a process without a work exchanger).</li> <li>2. Friction pressure loss of added piping needed to include the flow work exchanger in the reverse osmosis system. The plant supply pipe and filter loss are not included.</li> </ol>
$W_h$	Power input to high pressure brine pump	<ol style="list-style-type: none"> <li>1. Friction losses in pipes, check valves and control valves during brine in flow and pressurized feed outflow.</li> <li>2. Filling and emptying losses at inlet and outlet ports of accumulators.</li> <li>3. Power input when control valve is switching and flow is momentarily stopped.</li> <li>4. Pressure losses through membrane system on high pressure brine side.</li> <li>5. Frictional resistance to piston movement.</li> <li>6. Pressurization losses due to compressibility of water and resultant flow of high pressure brine.</li> </ol>	<ol style="list-style-type: none"> <li>1. <math>\Delta P_h</math> - measured pressure loss from control valve inlet port to check valve discharge.</li> <li>2. <math>P_h</math> - brine pressure.</li> <li>3. <math>Q_h</math> - high pressure brine flow rate.</li> <li>4. Fraction of cycle corresponding to no-flow operation.</li> </ol>	<ol style="list-style-type: none"> <li>1. Friction pressure loss of piping system connecting the flow work exchanger to the reverse osmosis unit.</li> <li>2. Pressure drop on the brine side of the membrane system.</li> <li>3. Pressure loss across a flow rate control valve which may be necessary to adjust the flow work exchange system flow to permit operation at off-design flow rates. A pump by-pass control could also be used.</li> </ol>
$W_c$	Power input into control system components	<ol style="list-style-type: none"> <li>1. Power input into solenoids, flow switches, pressure switches, etc.</li> </ol>	<ol style="list-style-type: none"> <li>1. none</li> </ol>	<ol style="list-style-type: none"> <li>1. Power input will be calculated from the voltage and current ratings and from the fraction of time the power is used.</li> </ol>
$W_a$	Equivalent pneumatic power input for control valve activation	<ol style="list-style-type: none"> <li>1. Pressurized air is used to actuate the 4-way control valve.</li> </ol>	<ol style="list-style-type: none"> <li>1. Air supply pressure.</li> </ol>	<ol style="list-style-type: none"> <li>1. Air consumption per cycle can be estimated so that power required can be calculated.</li> </ol>

In terms of pressure losses and flow rates, the equation becomes as follows:

$$W_h = (1 - F_h) \frac{\Delta P_h Q_h}{\eta_p \eta_m} (.321) + F_h W_{th} \quad (A-6)$$

The terms  $\Delta P_\ell$  and  $\Delta P_h$  are the values measured on the two existing flow work exchanger test systems. However, as shown on Table A-1, the measured  $\Delta P$  terms do not include all of the losses that should be accounted for when a flow work exchanger is included in a reverse osmosis system. Each of the pumps would require additional head capacity at the plant operating condition in order to provide for the additional losses listed in the right-hand column of Table A-1. In some cases these additional losses may be substantial and therefore must be evaluated to obtain a true measure of the flow work exchanger efficiency. The flow work exchanger efficiency should not be penalized for the pressure drop through the salt water side of the membrane system even though the pressure drop must be supplied by the high pressure, low head pump. This pressure drop must be considered when selecting this pump and evaluating the cost of production of fresh water.

When the flow work exchanger performance is defined in terms of power input values, the motor input power to the high pressure pump includes the effect of the surge of high pressure brine during pressurization of the feed stream.

The control and pneumatic power input terms will be a small percentage of the total power input, especially for the larger flow rate and higher pressure flow work exchangers.

In OSW report No. 357 (Ref. 1), the efficiency of the flow work exchanger is defined in terms of areas on the non-ideal indicator diagram shown on Figure A-2. This diagram shows the pressures that exist at the test system pressure measurement locations and not the pressures in the pressure vessels. The same four processes shown on Figure A-1 are exhibited in their non-ideal form on Figure A-2.

The losses associated with the low pressure feed inflow and high pressure feed outflow are due to the pressure differences needed to overcome pressure losses (3-4-9-10 and 1-2-6'-7). The ideal pressure vessel pressures on Figure A-1 are located inside the areas designated above. The lost work per cycle represented by these areas is:

$$(P_{\ell_1} - P_{\ell_2}) (V_4 - V_3) = \Delta P_{\ell} (V_4 - V_3) \quad (A-7)$$

$$(P_{h_2} - P_{h_1}) (V_1 - V_2) = \Delta P_h (V_1 - V_2) \quad (A-8)$$

These values multiplied by  $(2N)(144)(1-F)$  and divided by the pump and motor efficiencies result in values equivalent to the steady flow terms of the power loss equations, (A-4) and (A-5).  $N$  is the cycling rate and  $(1 - F)$  is the portion of the cycle time at the normal operating condition.

The pressurization and depressurization portions of the cycle are transient operating conditions that occur when the valve switches. During the pressurization of the feed flow, the brine experiences a sudden surge of flow as the valve opens to allow brine to flow into the cylinder. The brine pressure drops from point 6 and then quickly builds up to point 6' pressurizing the feed flow (4 to 1) as a small amount of brine flows into the cylinder and moves the piston slightly (5 to 1). This process produces a loss in the high pressure system represented by (4-6-6'-1) on Figure A-2 where the process from 4 to 1 is approximated by a straight line. This loss represents the difference between the energy available (6-6'-12-B) in the high pressure brine and the work delivered in pressurizing the low pressure feed (4-1-12-B). These losses combined with the transient operation during valve switching are included in the second term of the high pressure pump power input defined in Equation (A-6).

The high pressure pump experiences a drop in flow as the control valve switches, then a surge of flow during pressurization. The input motor power will also experience similar fluctuations during this transient period. Since this transient period is very short, the motor power input is dominated by the steady flow operating point.

The depressurization part of the cycle occurs with no flow passing through the low pressure pump since the check valve will not open until the pressure in the pressure vessel drops below the low pressure pump discharge pressure. The pump is operating at a zero-flow condition but a finite amount of time is needed for the shut-off head to build up after the flow stops. This part of the cycle is also transient operation. The second term of the low pressure pump input power given in Equation (A-4) describes the transient loss during valve switching and depressurization. Since the pump flow is stopped during the depressurization, the area (7-8-9) is not equivalent to the transient input power. Instead, it represents a surge of flow out of the exhaust line but it does not place any additional load on the low pressure pump.

The evaluation of the Flow Work Exchanger efficiency in terms of power inputs is the more meaningful of the two methods of description. Since input power in the flow work exchanger is the operating cost to be paid for, it is this power which should be used for system performance evaluation. This method of performance evaluation is equivalent to the method presented in OSW report No. 357 (Ref. 1).

### A.3 System Efficiency Losses Due to Leakage of High Pressure Flow

Leakage losses will decrease the Flow Work Exchanger efficiency by permitting water at high pressure to leak past check valves, pistons and control valves. The leakage flows represent energy losses in the high pressure brine flow from the reverse osmosis system or in the high pressure feed flow back to the reverse osmosis system. The leakage of high pressure brine flow past the closed exhaust control ball valve represents energy lost from the system. The leakage past the closed check valve represents a reduction of the high pressure feed flow to the reverse osmosis system and also a reduction of pressure level because of the throttling process. Leakage past the pistons or through the bladder changes the salt concentration of the high pressure feed flow and causes increased Flow Work Exchange system losses on the low pressure side of the system. Including the effect of leakage losses results in the following equation for Flow Work Exchanger Efficiency:

$$\eta = 1 - \frac{W_l/\eta_{v1} + W_h/\eta_{v2} + W_c + W_a}{W_i \eta_{v2}} \quad (A-9)$$

where the volumetric efficiency is defined as follows:

$$\eta_v = \frac{\text{Flow rate without leakage}}{\text{Flow rate with leakage}} \quad (\text{A-10})$$

$\eta_{v1}$  includes the effect of leakage past the piston and leakage past the check valve from the high pressure region.  $\eta_{v2}$  includes the effect of high pressure valve leakage. If measured values of flow rate are used to calculate  $W_l$ ,  $W_h$ , and  $W_i$ , then the effect of leakage has already been included and the volumetric efficiency correction is not needed. If the flow rates are calculated from piston displacement and cycle frequency, then the volumetric efficiency correction is required.

Appendix B  
DETAILED EVALUATION OF COST AND EFFICIENCY FOR A  
RANGE OF FLOW WORK EXCHANGER SIZES

**B.1 Description of Approach**

To evaluate the scale-up potential and estimate the cost and efficiency of flow work exchangers of different sizes, detailed component information was obtained. This information included size range, cost, and pressure losses for equipment suitable for use with salt water at pressure levels up to 1500 psig.

To carry out the analysis, a "standard system" was selected which includes all of the important system components of a flow work exchanger. Figure B-1 shows a schematic of this system with the components identified. Information has been obtained on each component in sizes suitable for use with accumulators of from 10 to 80 gallons capacity. The components are listed in Table B-1 together with specifications and potential suppliers.

The maximum flow rate capacity for an accumulator of given size is assumed to be 95% of the capacity corresponding to a cycling rate of 5 complete cycles per minute. This means that the control valves switch every six seconds and the flow rate in gpm is given by the following equation:

$$Q_{\max} = .95 V (\text{cpm}) 2 \quad (\text{B.1})$$

or

$$Q_{\max} = 9.5 V$$

V gal.	Brine GPM	Brine GPD	Approximate Reverse Osmosis Plant Fresh Water Production (based on 35% recovery from feedwater) GPD
2	19.0	27,500	14,800
10	95	135,000	72,700
20	190	275,000	148,000
40	380	550,000	296,000
80	760	1,100,000	592,000

**Figure B-1**  
**Standard Flow Work Exchanger for Cost Effective Analysis**

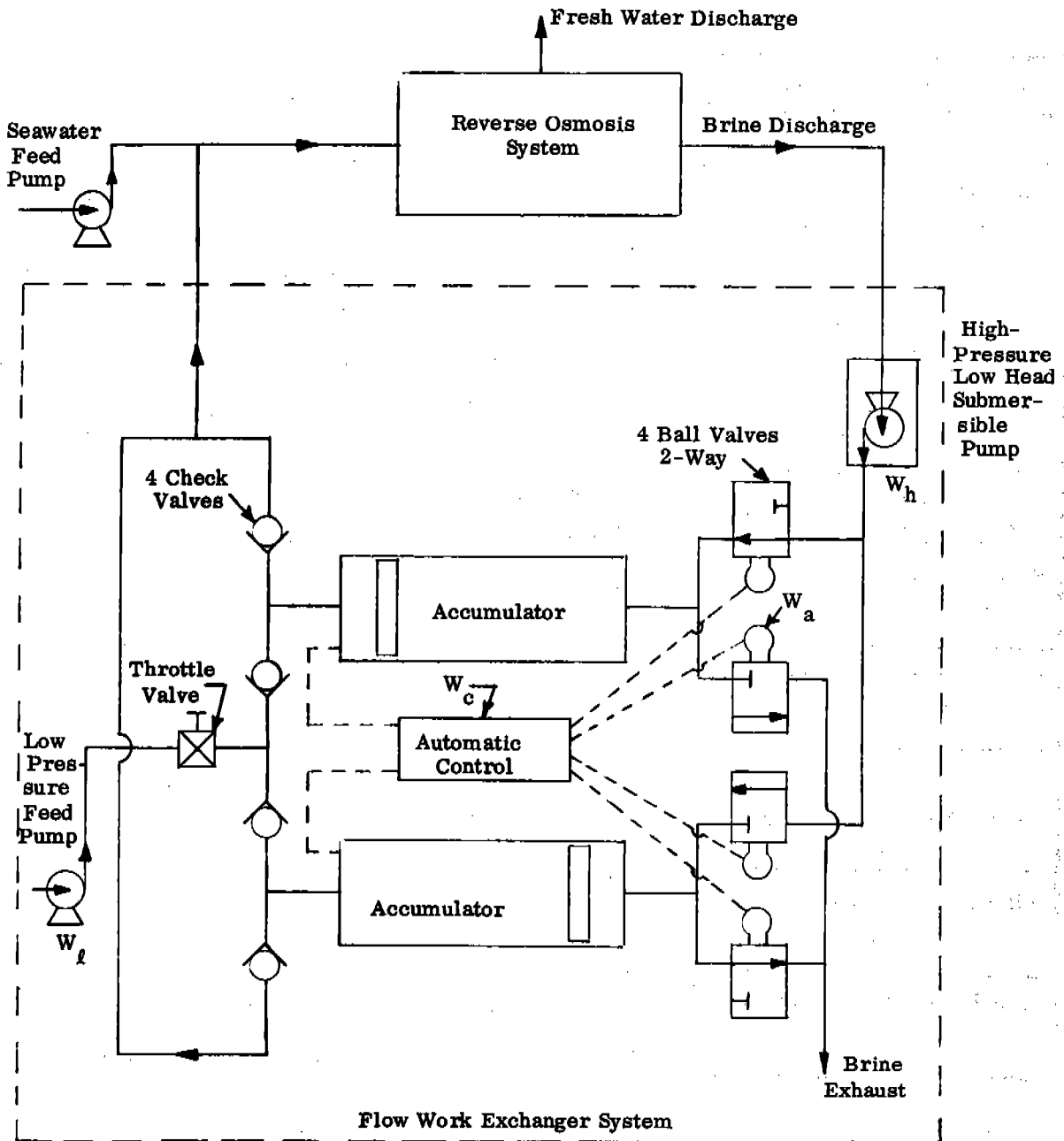


Table B-1

## COMPONENTS FOR STANDARD SYSTEM IN FIGURE B-1

Standard System Components	Specifications	Potential Supplier
2 Accumulators for Water Service	1500 psi 100° F salt water	<u>Piston Type</u> 1. Superior Hydraulics 2. Greer Olaer Products 3. Liquidonics, Inc. <u>Bladder Type</u> 1. Greer Olaer Products 2. Hydrotrole, Ltd.
1 Low-Pressure-Low-Head Pump	60 psi 100° F salt water	1. Goulds Pumps, Inc. 2. Decatur Pump Co. 3. Crane Co.
1 High-Pressure-Low-Head Pump (submersible) with Pressure Vessel	1500 psi 100° F salt water	1. Reda Pump Company 2. Pleuger Submersible Pumps, Inc.
4 Check Valves	1500 psi 100° F salt water	1. Circle Seal Co. 2. Mueller Steam Specialty Co. 3. Combination Pump Valve Co.
4 Ball Valves with Solenoid or Pneumatic Actuation	1500 psi 100° F salt water	1. Fluid Dynamics Inc. 2. Jamesbury Corporation 3. ebv Systems Inc.
Automatic Control System		
Piping and Fittings	1500 psi 100° F salt water	
Low Pressure Throttle Valve	60 psi 100° F salt water	

The following sections present information on the components needed in the flow work exchange system. Several combinations of accumulators and components are then evaluated in order to estimate the system cost and efficiency as a function of size.

## B.2 Component Pressure Loss Information

Pressure loss occurs in all of the following components in both the high-pressure and low-pressure portions of the system.

1. check valves
2. ball valves
3. expansion and contraction losses from accumulator to pipe
4. pipe friction and elbow losses
5. pipe expansions and contractions to accommodate check valves and ball valves of different sizes
6. low-pressure throttle valve

The values for pressure loss as a function of flow rate for various sizes of ball valves and check valves were obtained from manufacturers' catalogs and are presented in Figures B-2 and B-3.

The pressure losses in contractions, expansions, pipes, and fittings were calculated using information from references 2 and 3. The accumulator entrance and discharge pressure losses are caused by expansion and contraction of the fluid at the accumulator ports. The following equations were used to evaluate these losses:

$$\text{expansion loss } H_e = \frac{C_1^2}{2g} \left(1 - \frac{A_1}{A_2}\right)^2 \quad (\text{B.2})$$

$$\text{contraction loss } H_{co} = \frac{C_3^2}{2g} K \quad (\text{B.3})$$

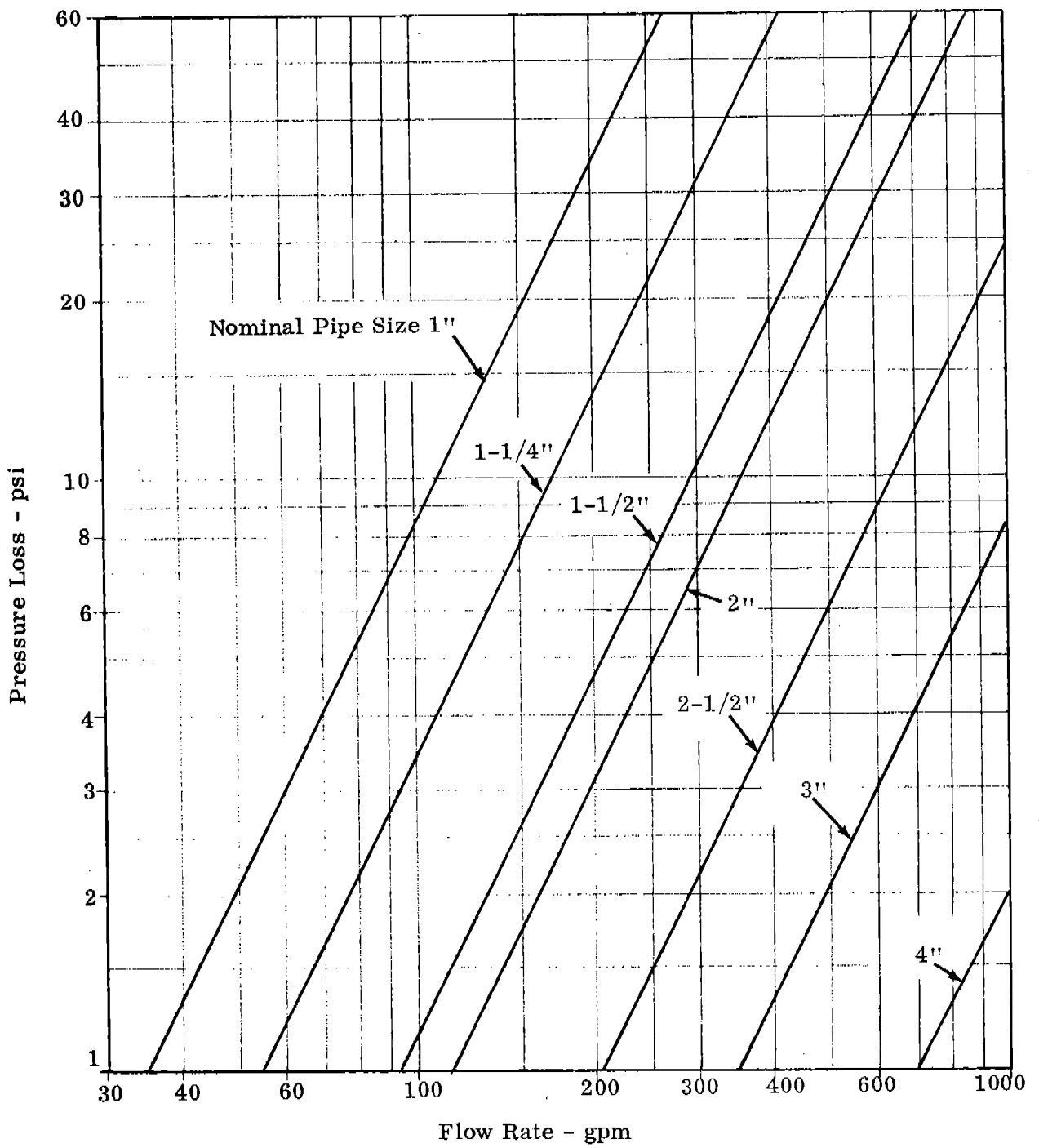


Figure B-2. Ball Valve Pressure Loss

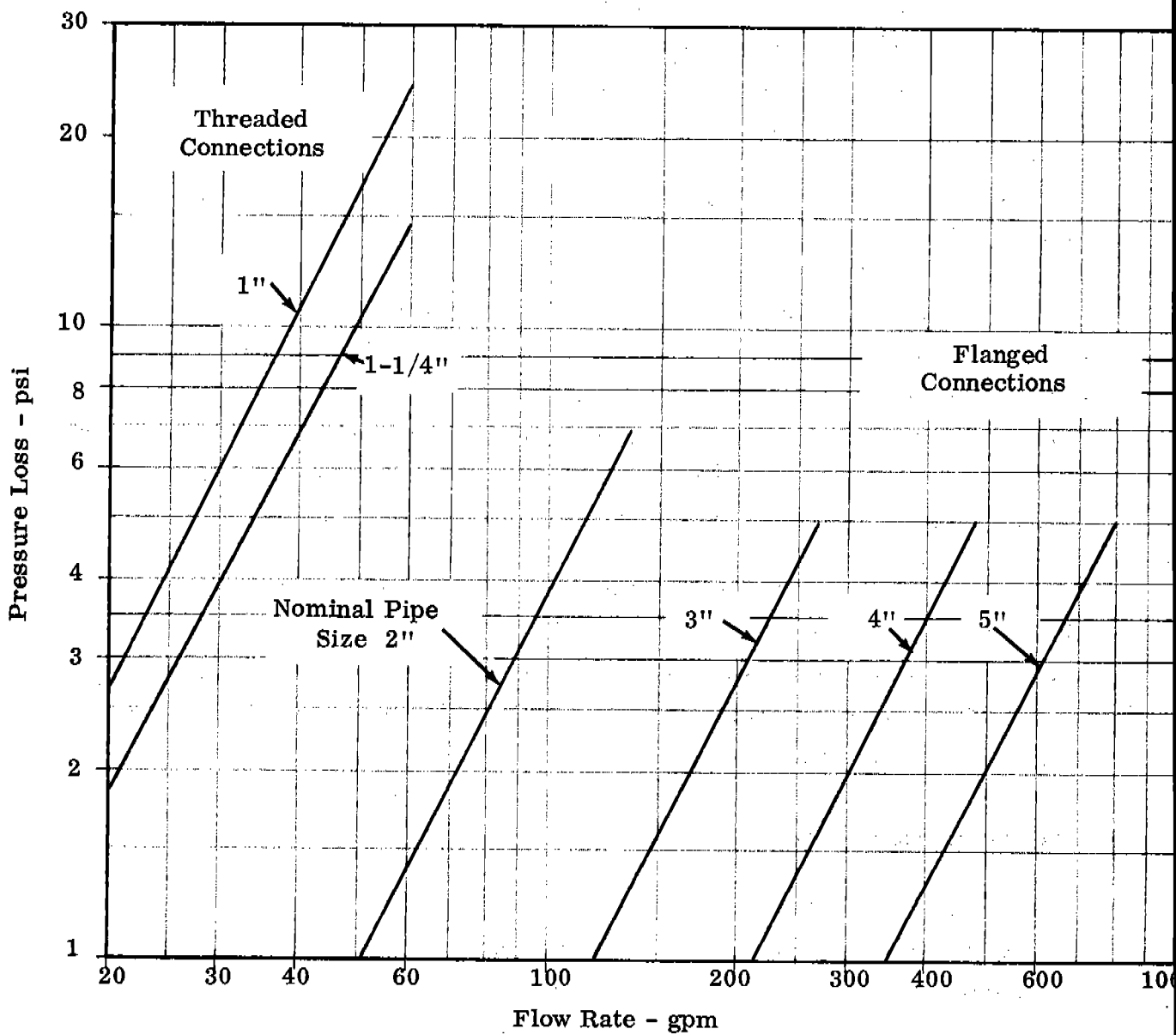


Figure B-3. Check Valve Pressure Loss

where

$C_1$	- fps	-	velocity in inlet pipe
$C_3$	- fps	-	velocity in discharge pipe
$A_1$	- in <sup>2</sup>	-	area of inlet pipe
$A_2$	- in <sup>2</sup>	-	area of accumulator
K		-	loss coefficient from Figure 4 of Reference 2

The combined expansion and contraction loss is presented in Figure B-4 for different accumulator sizes and port sizes.

To allow estimation of the pressure loss in system piping and fittings, the flow path for both the high pressure and low pressure lines was assumed to consist of 10 feet of pipe and six 90° bends. To calculate an equivalent length of pipe, each 90° pipe bend was assumed to have a loss equivalent to 35 diameters of straight pipe. Figures 7 to 14 of Reference 2 were used to calculate the pressure loss information presented in Figure B-5 for different size pipes.

Some combinations of piping and valves will require additional expansions and contractions in pipe sizes. These combinations were evaluated in a manner similar to that used for the accumulator entrance and discharge losses. The throttle valve losses in the low pressure line were assumed to be 0.15 pipe velocity heads for a straight through-flow valve (i.e., gate valve).

### B.3 Component Cost Data

Figure B-6 shows component costs as a function of size for accumulators, ball valves, and check valves. This information was obtained from a number of manufacturers.

The bladder and piston accumulator prices are for hydro-pneumatic accumulators designed for standard water service. The accumulator vessels are made of carbon steel with an internal phenolic coating or chrome plating. The port and internal components are stainless steel and the piston is anodized aluminum. The piston-type accumulators have standard ports as large as 3 inches in diameter. The bladder type units have a 2" to 4" port on one end and a small gas port attached directly to the

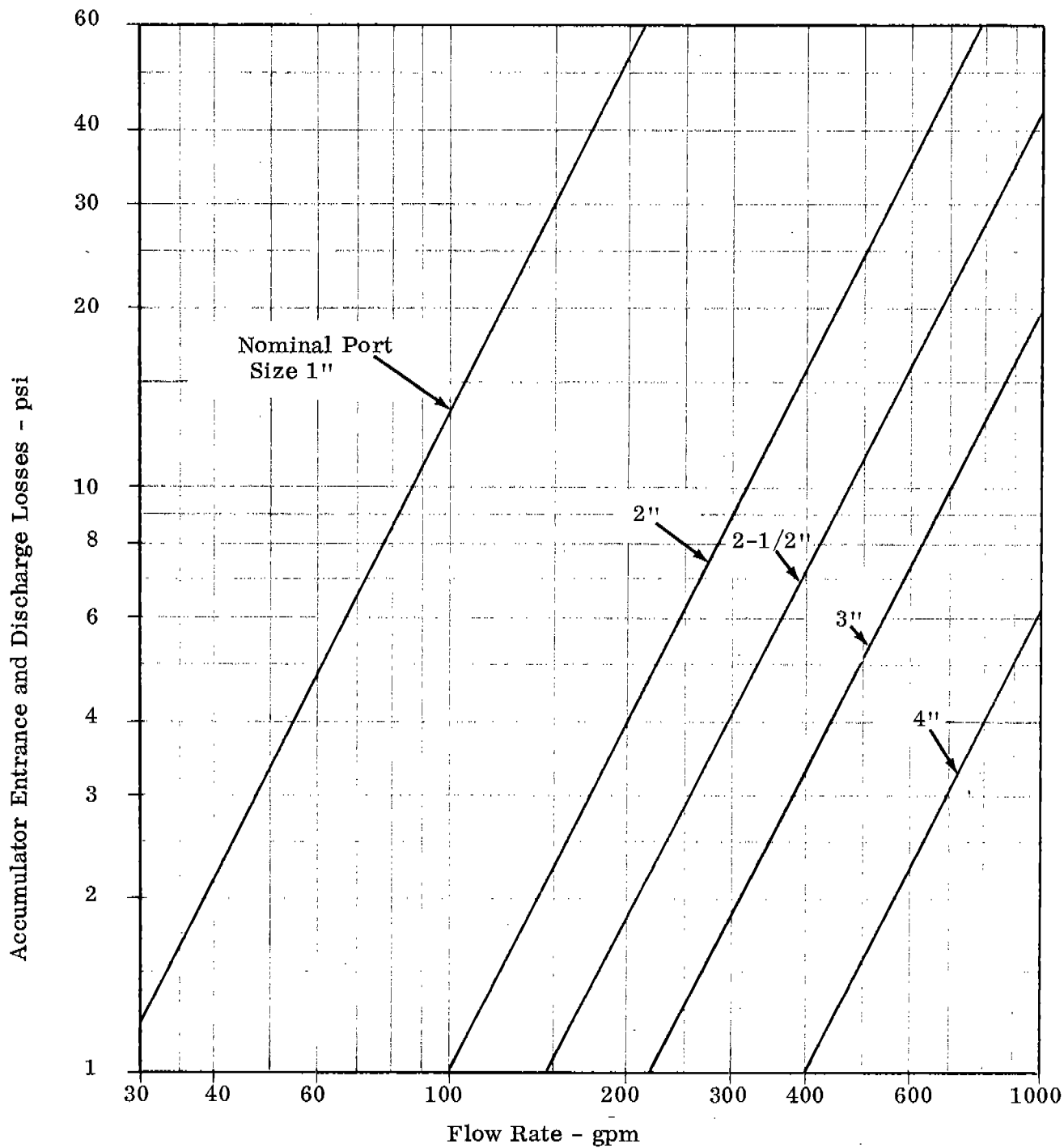


Figure B-4. Piston Accumulator Entrance and Discharge Losses Due to Expansions and Contractions

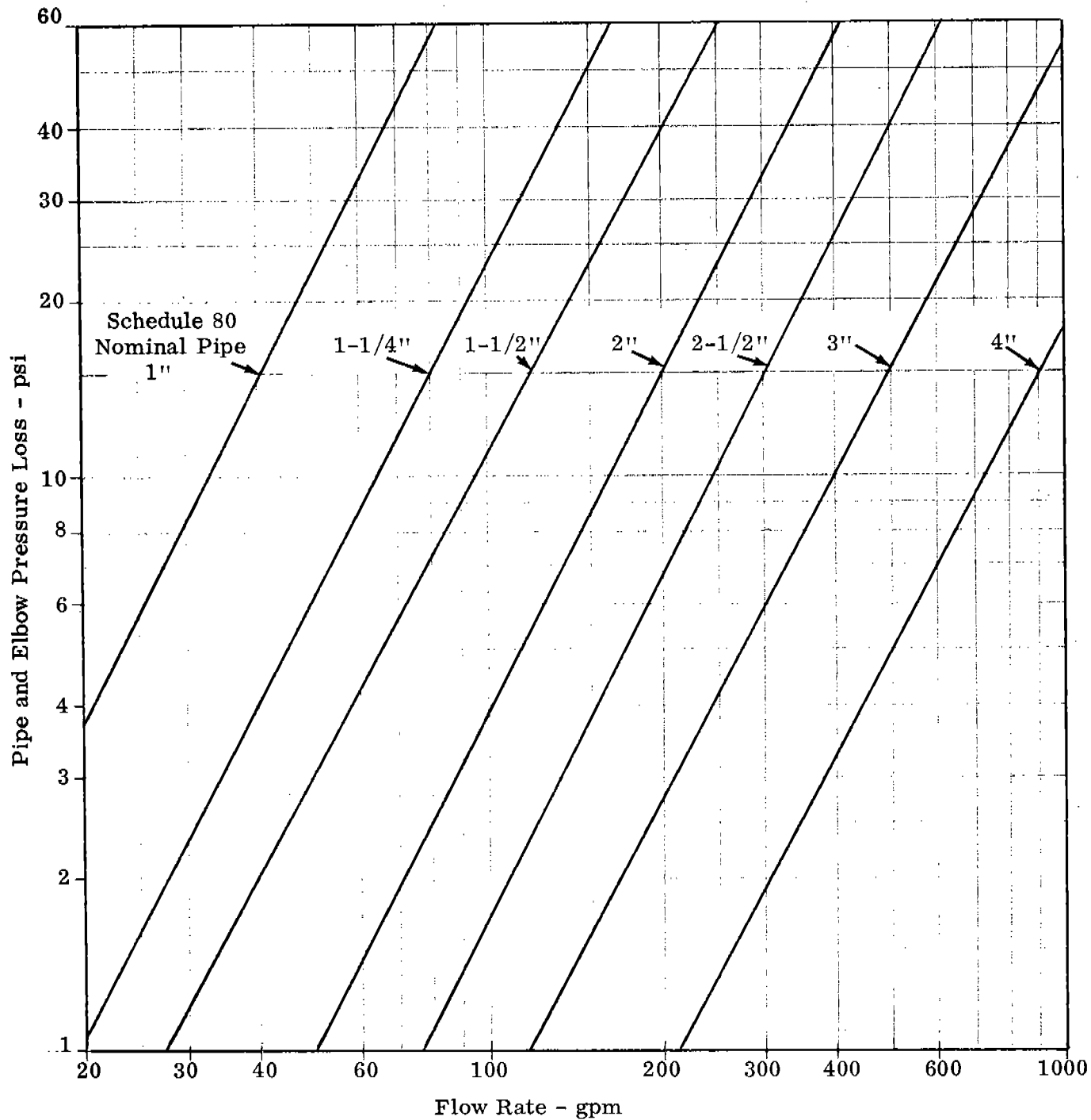


Figure B-5. Pipe and Elbow Pressure Loss  
10 ft. of Pipe with 6 Elbows

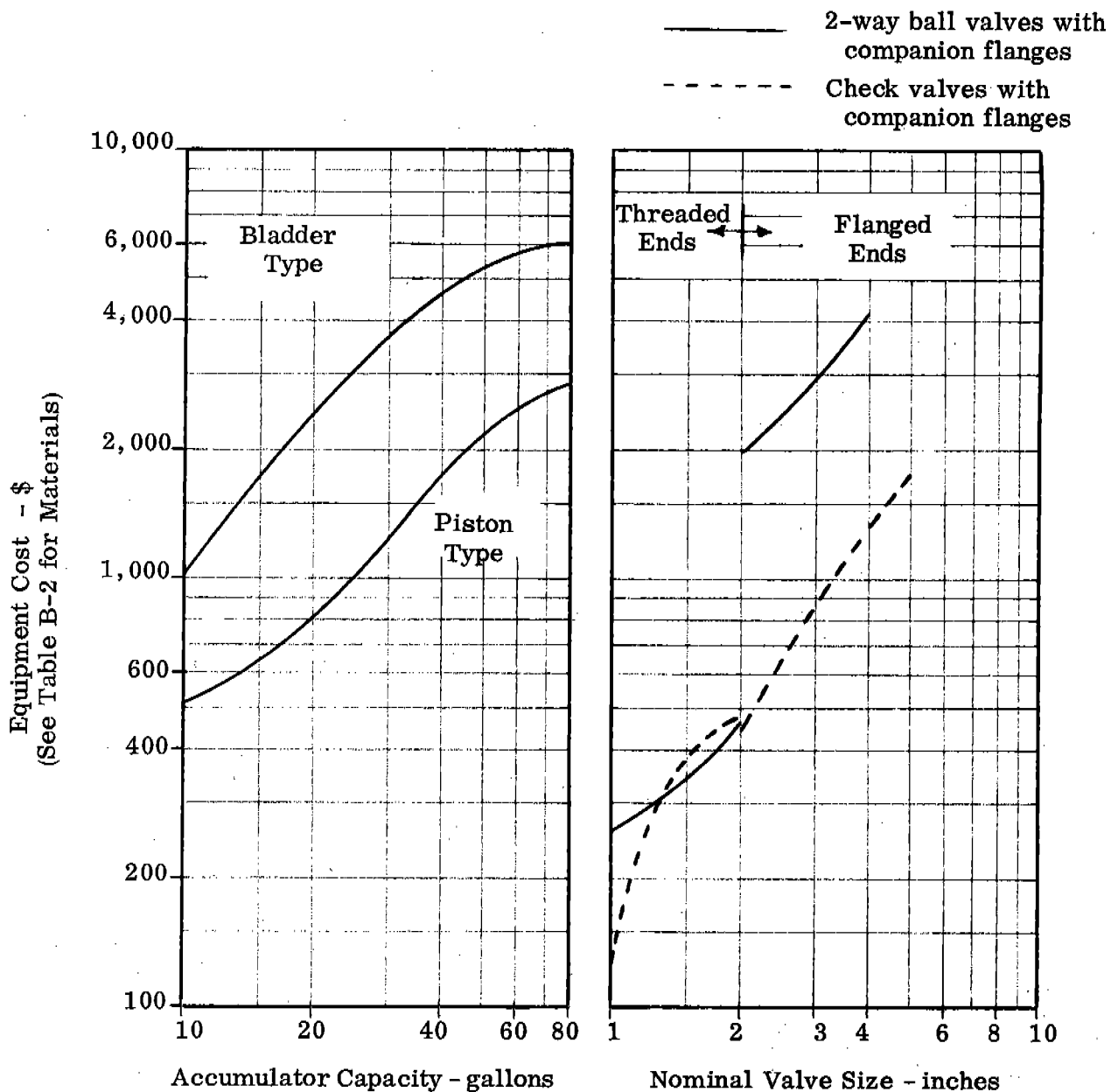


Figure B-6. Cost of Accumulators, Ball Valves, and Check Valves

bladder (approximately 5/8" inner diameter) on the other end. The bladder units are more expensive and, for work exchanger service, would require modifications of the bladder and vessel to install a larger port on the bladder end. Accumulator shells can be made from corrosion resistant materials such as Monel at significantly higher cost (4 times).

The ball valves and check valves are both made of stainless steel. The ball valve prices include a solenoid or pneumatic actuator and a set of mating flanges as required. The larger check valves are of the wafer type and the price includes the required mating flanges. A large increase in price results for the larger size components when flanges are required instead of threaded connections.

Figure B-7 shows the cost of low-pressure, low-head pumps with motors. It also shows costs for high-pressure, low-head submersible pumps. The low-pressure pump costs are shown for two materials: bronze and stainless steel. This information was obtained from the manufacturers and from Reference 4. The high-pressure submersible pump costs were obtained from manufacturers and apply for the nickel alloy material which they recommend for brine applications. Cost data for both types of pumps are plotted against the parameter used in Reference 4; gpm x psi.

Reference 4 was used to obtain prices for pipe, 90° elbows, tees, flanges, and gate valves. A check of the cost curves presented in this reference showed that most of the information given for schedule 80 and 1500 psi-rated stainless steel components remains reasonably accurate. However, local prices for schedule 80 stainless steel pipe were found to be higher by a factor of 3. Therefore, the Reference 4 pipe cost data was increased by a factor of 3 for use in this study.

#### B.4 Pump Efficiency Data

Pump efficiency values are important factors in calculating the efficiency of the flow work exchanger. To obtain efficiency values for the low-pressure, low-head pumps and for the high-pressure, low-head submersible pumps, Reference 5 and manufacturers' data were used. Figure B-8 shows the efficiency values for both classes of pumps as determined from both of these sources.

- Low-pressure, low-head-rise pump
- - - - High-pressure, low-head-rise submersible pump (nickel alloy)

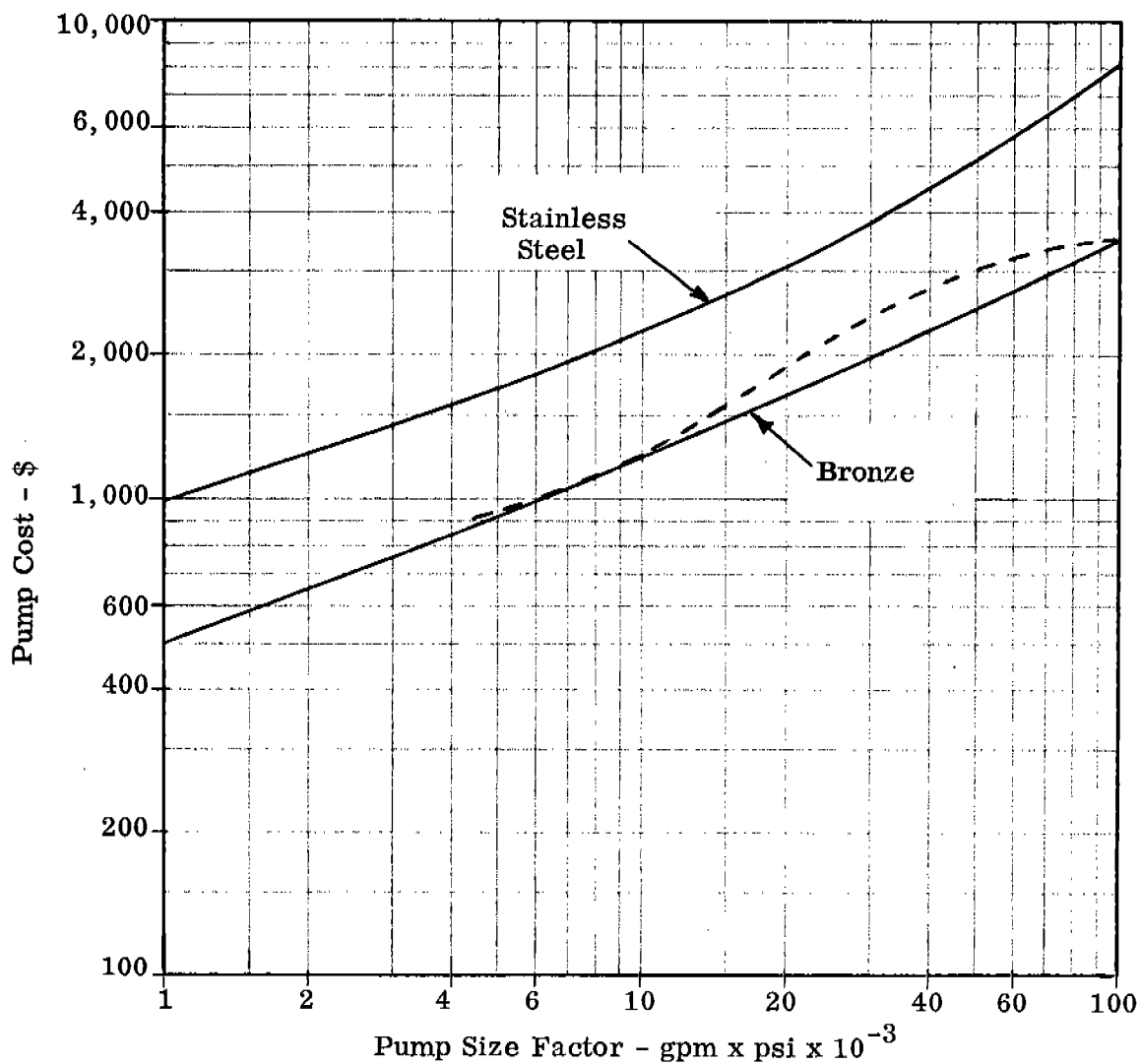


Figure B-7. Flow Work Exchanger Pump Costs

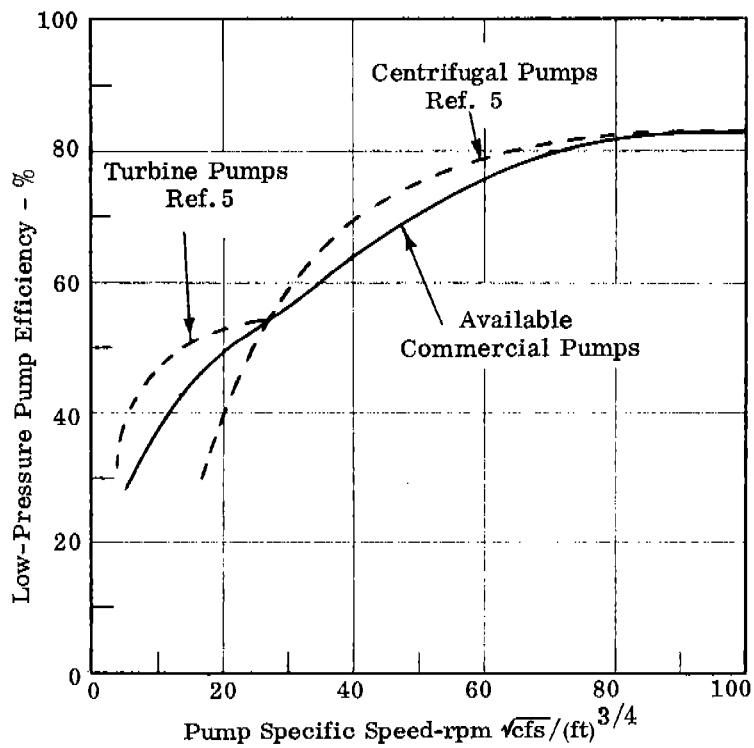
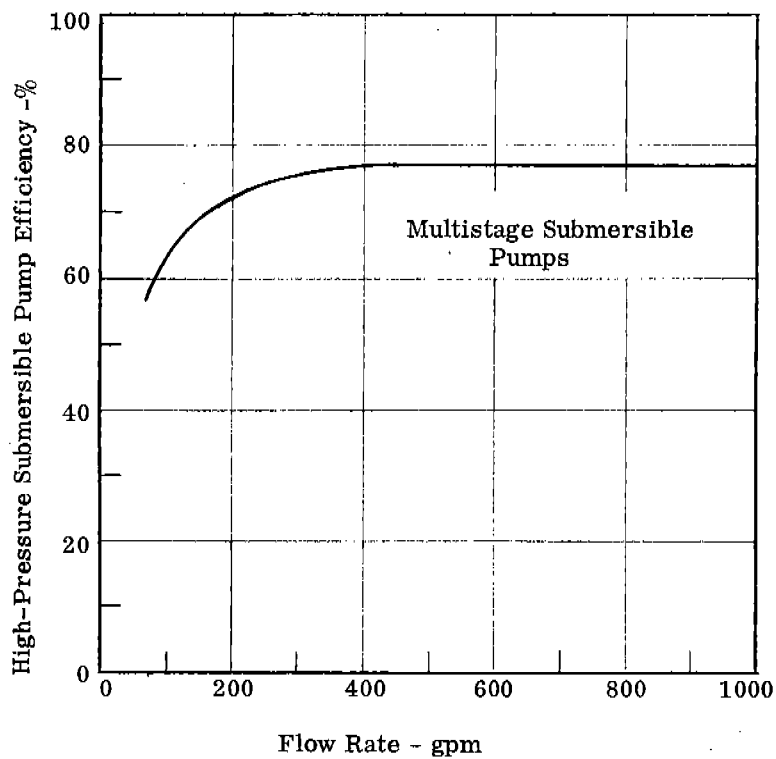


Figure B-8. Pump Efficiency Data

The high-pressure, low-head submersible pump efficiency values represent the peak efficiency envelope of a series of pumps of different sizes. Each individual pump will only operate over a small range of flow (about  $\pm 20\%$ ) with efficiencies close to this curve. Operating efficiencies of these pumps outside this operating range drop rapidly.

The low-pressure, low-head pump efficiency is shown on Figure B-8 as taken from a general correlation in Reference 5 for single-stage centrifugal pumps and turbine-type pumps. The centrifugal pumps produce the highest efficiency levels in the specific speed range above 30. This generally corresponds to flow rates above 50 gpm for low-head-rise pumps. The turbine-type pump provides the highest efficiency for specific speeds and flow rates less than the above values. Manufacturers' data for both types of pumps were plotted on Figure B-6 and the upper envelope of this commercial data was drawn as the solid line. This envelope will be used for evaluating the efficiency of larger-size flow work exchangers.

#### B.5 Selection and Evaluation of 18 Flow Work Exchanger Systems

To evaluate the cost and efficiency of large-size flow work exchangers, a series of 18 systems were sized and analyzed using the component information in Sections B.2, B.3, and B.4. Four accumulator sizes (10, 20, 40, and 80 gallons) were chosen and the other required components were matched and selected in order to achieve an operating efficiency of about 91% at the maximum flow rates listed in Section B.1. Each of the four systems were analyzed at 50%, 75%, and 100% of maximum flow. This corresponds to a fresh water production of approximately 35,000 to 600,000 gpd from a reverse osmosis plant operating with a recovery factor of 35%. Larger capacities can be attained by adding several smaller-sized units in parallel. In addition to the 12 units described above, an additional 6 units using the 40 gallon accumulator were sized and analyzed to determine the effect of increased and decreased levels of pressure loss on the unit cost and efficiency.

Table B-2 shows the selected component sizes and costs for each unit analyzed. Table B-3 shows the pressure losses for each unit at three flow rates. Each of the 18 units are considered to be operating at design conditions so different pumps are selected for each of the 18 units to supply the heads and flows listed on Table B-3. The cost and efficiency results are shown on Figure 6-1.

The following assumptions were made to prepare the results presented in Tables B-2 and B-3.

1. The pumps are assumed to be operating at the peak efficiencies shown on Figure B-8 when supplying the flow rates and heads listed on Table B-3.
2. The power supplied to solenoids and pneumatic controls is assumed to be negligible.
3. The motor efficiencies are assumed to be 80%.
4. The pump power inputs during the transient valve switching period are assumed to be equal to the steady-state power inputs.
5. The transient time period is assumed to be short compared to the cycle time.
6. The volumetric efficiency is assumed to be 100%. As piston rings and valve seats wear, leakage will reduce the flow work exchanger efficiency values according to Eq. (A-9).

Pressure losses in addition to those listed on Table B-3 may occur. These will increase the required pump heads and equipment cost. Such additional losses can be caused by the following components.

1. Friction losses due to the piston seals or pressure losses due to the stretching of a bladder during part of its cycle.
2. Pressure losses in the high pressure lines connecting the brine and feedwater sides of the work exchanger unit to a reverse osmosis plant. The connections may require hundreds of feet of pipe and thus involve significant pressure losses.

Table B-2

## TABULATION OF COMPONENT SIZES AND COSTS FOR COST EFFECTIVE ANALYSIS

Components	Unit 1		Unit 2		Unit 3A		Unit 3B		Unit 3C		Unit 4		Materials
	Size	\$	Size	\$	Size	\$	Size	\$	Size	\$	Size	\$	
2 Piston Type Accumulators	10 gal. 2" port	1040.0	20 gal. 2" port	1560.0	40-gal. 2" port	3400.0	40-gal. 2-1/2" port	3400.0	40-gal. 3" port	3400.0	80-gal. 3" port	5800.0	Carbon steel tube with phenolic or chrome coating. Port assemblies stainless steel.
4 Ball Valves with companion flanges	1" threaded	1040.0	1-1/2" threaded	1360.0	2" threaded	1800.0	2-1/2" flanged	9600.0	3" flanged	11600.0	3" flanged	11600.0	316 stainless steel
4 Check Valves with companion flanges	2"	1920.0	3"	3360.0	4"	5200.0	4"	5200.0	4"	5200.0	5"	7000.0	316 stainless steel
Low Pressure Pump		a. 690. b. 520. c. 320.		a. 960. b. 680. c. 475.		a. 2200. b. 1500. c. 875.		a. 1520. b. 1050. c. 640.		a. 1050. b. 750. c. 520.		a. 1950. b. 1350. c. 820.	Bronze
High Pressure Pump plus Pressure Vessel		a. 1340. b. 1220. c. 1040.		a. 1760. b. 1580. c. 1420.		a. 4350. b. 3200. c. 2620.		a. 3200. b. 2720. c. 2480.		a. 2720. b. 2560. c. 2400.		a. 5250. b. 4200. c. 3780.	Nickel alloy pump
Low Pressure Gate Valve with Companion Flanges	2" flanged	464.	2" flanged	656.	2" flanged	876.	2-1/2" flanged	876.	3" flanged	876.	4" flanged	1096.	316 stainless steel
20' Pipe, 6 Elbows (weld), 2 flanges 8 Tees (weld)	2"	1540.	2"	1540.	2"	1540.	2-1/2"	2070.	3"	2720.	4"	4000.	316 stainless steel
Expansions and Contractions (weld)	2 to 1 1 to 2	16.	2 to 3 3 to 2 2 to 1-1/2 1-1/2 to 2	40.	2 to 4 4 to 2	36.	2-1/2 to 4 4 to 2-1/2	36.	3 to 4 4 to 3	36.	3 to 4 3 to 5 5 to 4	96.	316 stainless steel
Automatic Control		500.		500.		500.		500.		500.		500.	
Miscellaneous Parts (20% of total cost)		1710.		2350.		4000.		5300.		5600.		7500.	
Total Unit Cost		a. 10,260 b. 9,970 c. 9,590		a. 14,090 b. 13,630 c. 13,260		a. 23,900 b. 22,052 c. 20,850		a. 31,700 b. 30,750 c. 30,100		a. 33,700 b. 31,250 c. 32,850		a. 44,800 b. 43,150 c. 42,200	

Table B-3  
FLOW WORK EXCHANGER EFFICIENCY

COMPONENT PRESSURE LOSS psi	UNIT 1			UNIT 2			UNIT 3A			UNIT 3B			UNIT 3C			UNIT 4		
	a	b	c	a	b	c	a	b	c	a	b	c	a	b	c	a	b	c
Accumulator Entrance and Exit	1.0	1.0	1.0	3.7	2.1	1.0	14.5	8.20	3.7	6.6	3.7	1.65	3.0	1.7	1.0	11.5	6.6	3.0
Ball Valve	7.4	4.2	1.9	4.2	2.3	1.0	11.5	6.40	2.9	3.5	1.90	1.0	1.4	1.0	1.0	4.2	2.7	1.2
Check Valve	3.4	1.9	1.0	2.5	1.4	1.0	3.2	2.80	1.0	3.2	2.8	1.0	3.2	2.8	1.0	4.5	2.5	1.2
10' Pipe Friction Plus Six 90° Bends	3.4	2.0	1.0	14.0	7.9	3.5	51.0	29.5	13.5	23.5	13.5	6.0	9.4	5.4	2.5	10.1	6.2	3.0
Contractions and Expansion	11.4	6.4	2.8	5.0	3.3	1.5	10.6	6.0	2.7	4.0	2.3	1.1	0.9	0.2	0.2	6.0	3.4	1.5
Gate Valve	.11	.06	.03	.43	.24	.11	1.72	.97	.43	.83	.47	.21	.34	.19	.08	.45	.26	.11
Total Loss-Low Pressure Line	26.7	15.6	7.8	29.8	17.2	8.1	92.5	53.9	24.2	41.6	24.7	11.0	18.2	11.3	5.8	36.8	21.7	10.0
Total Loss - High Pressure Line	26.6	15.5	7.7	29.4	17.0	8.0	90.8	52.9	23.8	40.8	24.2	10.8	17.9	11.2	5.7	36.3	21.4	9.9
Flow Rate GPM	95	71.25	47.5	190	142.5	95	380	285	190	380	285	190	380	285	190	760	570	380
Low Pressure Pump $\eta$	62%	70%	76%	69%	82%	83%	60%	62%	74%	72%	80%	83%	83%	83%	83%	82%	83%	83%
High Pressure Pump $\eta$	62%	59%	55%	71%	67%	62%	77%	76%	71%	77%	76%	71%	77%	76%	71%	77%	77%	77%
Power Input to Low Pressure Pump ft lb/sec	1,641.5	637.1	195.6	3,292.6	1,199.3	372.0	23,506.5	9,941.6	2,493.2	8,809.7	3,530.7	1,010.4	3,343.4	1,556.9	532.7	13,685.6	5,979.6	1,837.0
Power Input to High Pressure Pump ft lb/sec	1,635.4	751.1	266.8	3,156.9	1,450.8	491.9	17,980.2	7,959.7	2,555.5	8,079.1	3,641.3	1,159.6	3,544.5	1,685.3	558.4	14,376.2	6,356.4	1,960.4
Flow Work Exchanger Efficiency 1000 psi System	89.3%	93.9%	96.9%	89.4%	94.2%	97.2%	65.9%	80.4%	91.7%	86.1%	92.2%	96.4%	94.4%	96.3%	98.2%	88.3%	93.2%	96.9%
Flow Work Exchanger Efficiency 1500 psi System	92.8%	95.9%	97.9%	92.9%	96.1%	98.1%	77.3%	86.9%	94.4%	90.7%	94.8%	97.6%	96.2%	97.5%	98.8%	92.3%	95.5%	97.9%

3. Pressure losses on the brine side of the reverse osmosis membrane system.

Items 1 and 2 will contribute to the head requirements for the pumps and the cost of the required equipment. The effect of the added pressure loss should be included in the flow work exchanger efficiency. Item 3 above will contribute to the head requirements for the pumps and the pump cost. However, the work exchanger efficiency should not be penalized for this reverse osmosis system loss. Items 2 and 3 cannot be realistically included in the present analysis without consideration of the complete reverse osmosis system plant layout and operation. This is beyond the scope of this project. However, Figure B-9 does show the approximate effect of pressure losses on system efficiency and pump cost using unit (3Ba) as a base. Cost of the additional piping can be determined from Section B.3.

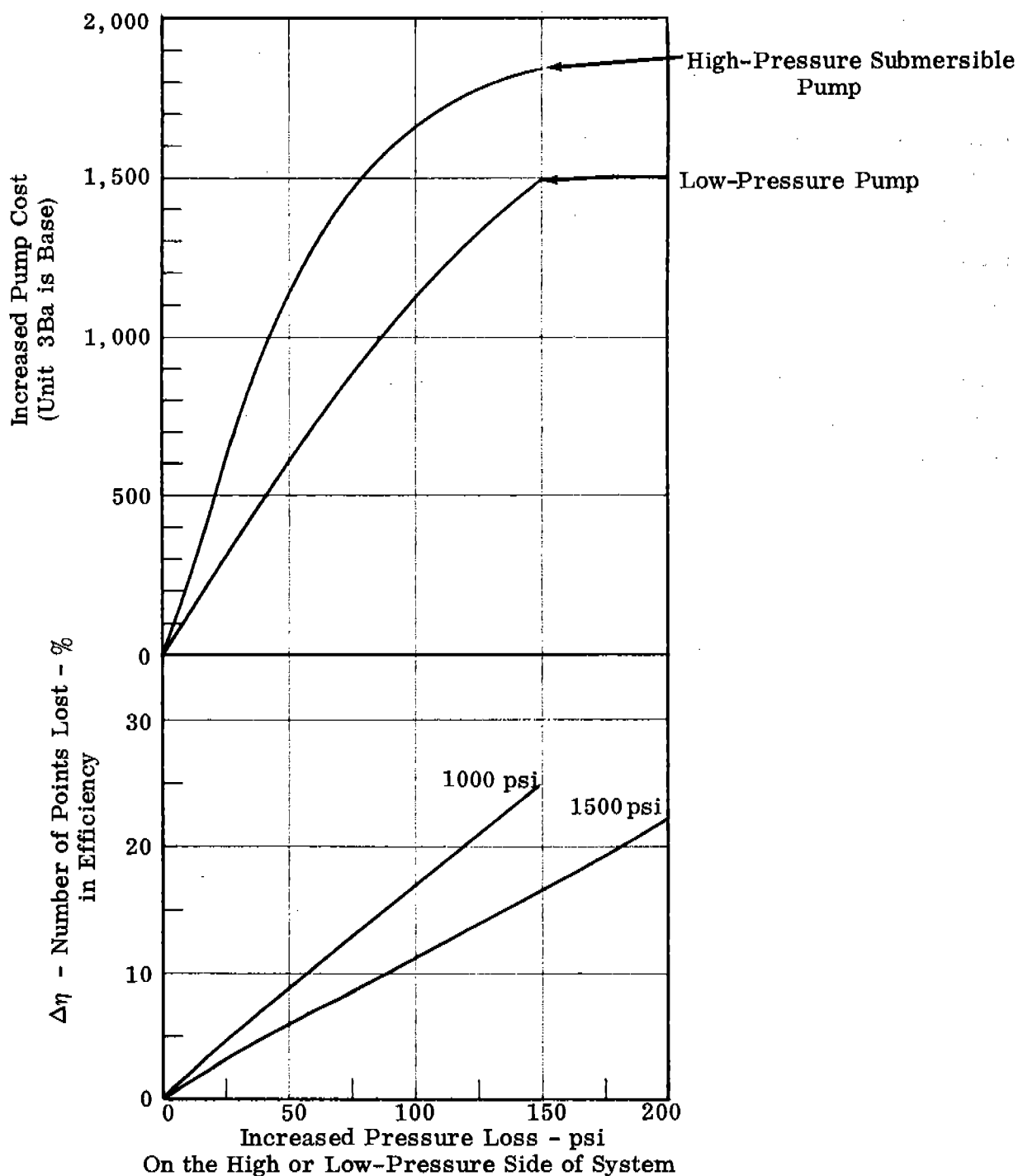


Figure B-9. Increased Pump Cost and Decreased Flow Work Exchanger Efficiency Due to Increased System Pressure Loss